The Association
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Engineering and Shipbuilding Draughtsmen

ELECTRIC TELPHERAGE.

By G. W. GROSSMITH, M.C., A.M.I.Mech.E.

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SESSION 1926-27.

Milne, Tannahill, & Methven, 12-14 Mill Street, Perth

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GENERAL.

In writing a short Paper under a heading of such a general nature as "Electric Telpherage," either of two alternative methods may be adopted. The paper may attempt to broadly cover the whole ground, briefly referring to a large variety of installations, describing the uses to which the telpherage system of mechanical handling may be applied, and showing how certain special handling problems have been solved by the use of telphers. The alternative method is to select examples which are fairly typical of the system, and deal with the design of these in closer detail than would be possible in a wider survey. Since this paper is written for a designer's Association, the second method will mainly be adopted, and in addition some of the more important general considerations affecting the design of all telpher structures will be briefly discussed.

The subject of automatic telpherage will not be referred to, as it involves almost totally different considerations, and in this paper the word telpher is to be understood to mean an electrically-driven machine, provided with hoisting and travelling motions, under the direct control of a driver who is carried in a cab or cage attached to the machine.

Readers not already familiar with the telpherage system of mechanical handling may, by reference to Fig. 1, gather some idea of the general appearance of a Telpher Transporter (in America called a man-trolley), and the single line overhead track upon which it runs, underhung from wheeled bogies.

Since the electric telpherage system was first developed in this country, about twenty-five years ago, it has found its chief application in gasworks and electricity stations, for the handling and storing of coal and coke in bulk, but it has also been much used for the handling of general merchandise, such as barrels, sacks of sugar (eight or ten in one sling at a time), and confectionery (by slings on wheeled trucks). Generally, a telpher is most useful in the conveying of coarse granular materials, such as coal, coke and ashes, and in these cases it carries either a skip-bucket or a grab, depending upon the means available for filling.

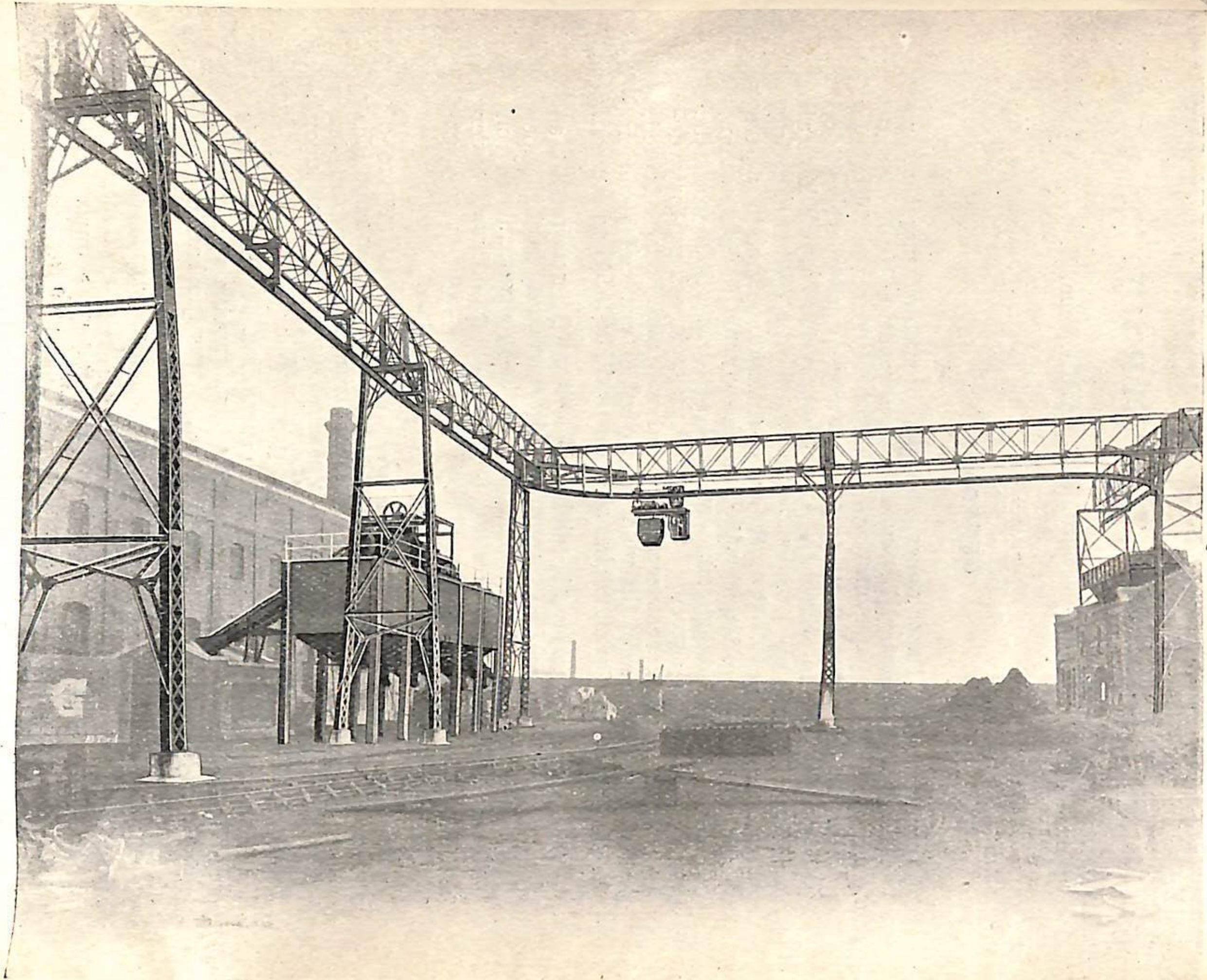

Fig. 1

The height and length of the track upon which the machine runs are governed entirely by the duty required of the plant, but there are economic limits to both. The telpher being a single-conveying unit, operating in a to-and-fro manner, has a lower capacity the longer the distance it has to travel, and it is unusual for a telpher to be occupied in conveying materials between points more than a thousand feet apart. There are many tracks much longer than a thousand feet, but in nearly all cases this is to accommodate a number of relatively short-distance operations, carried out alternately by one machine, or simultaneously by several machines operating on different sections of the track. The height usually ranges between forty and fifty feet, but some tracks are as low as twenty feet above ground level, when required for taking hot coke direct from a horizontal retort house which has a ground level bench, whilst plants which are required to discharge coal direct into the bunkers above the boilers of a large electricity generating station frequently require a track which is eighty feet and sometimes one hundred feet above ground level, and in these cases the supporting structure is very heavy and costly. The installation by Messrs. Robert Dempster at the Yallourn Power Station, Morwell, Australia, is a good example of a heavy-duty high-level telpher system. The track is of the bottom flange type, is 104 feet above ground level, and is arranged in four parallel lines converging by means of switches and turntables into two parallel lines over the coal bunkers. Three machines are at present in use, each carrying a coal grab and rated for a gross load of 8 tons.

Hoisting and travelling speeds do not vary very much. Average hoisting speeds are from sixty to eighty feet per minute, though machines on high tracks are usually arranged to hoist at speeds upwards of one hundred feet per minute. The travelling speed ranges from five hundred feet per minute to seven hundred feet per minute, the higher travelling speeds being provided when the track is fairly straight and long.

Telpher plants are sometimes installed in situations and for duties which could quite well be met by combined elevator and conveyor systems, or by gravity bucket conveyor systems. In these cases the advantages claimed for the telpher are that it has a comparatively small proportion of working parts, which keeps maintenance costs low ; that although a certain amount of energy is dissipated by lowering the empty skip or grab once in each working cycle, the overall mechanical efficiency is as great as, or greater than that of the continuous bucket or belt system ; also that the telpher works only when there is useful work to be done, whereas it is common for continuous systems to be kept working incessantly in situations where the supply is intermittent.

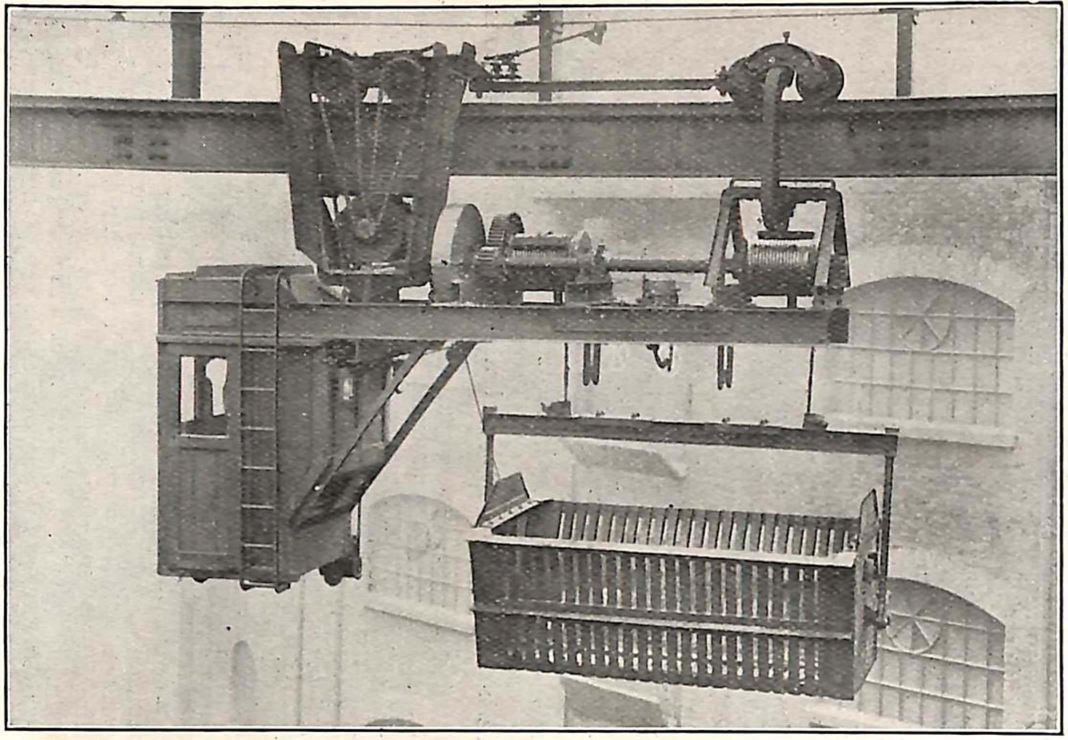

Fig. 2

THE TELPHER MACHINE.

The hoisting gear usually consists of a double reduction spur gear, driven by an independent electric motor. The hoisting barrel may be single or double. It is sometimes essential that the load should not spin, or even " screw " appreciably, when taking the curves in the track, and in such cases twin hoisting ropes well spaced apart are used. For tracks of moderate heights it is not usual to provide a clutch or sliding pinion, and lowering is done by the weight of the empty skip or grab revolving all the gears and the motor, under control of a brake. For high lifts this method would be too wasteful of time, since it is undesirable to revolve the motor at more than about twice its normal full load speed, owing to the centrifugal forces set up in the rotor (the writer has known A.C. motors to burst their rotors due to this cause), and in such cases the barrel is released by a sliding pinion and the load lowered very rapidly under the control of a band brake operating direct on to a brake drum attached to the barrel. In addition to a hand-brake, a series solenoid brake is usually provided, and this operates on a brake drum attached to the armature shaft or its extension. This electrical brake is in circuit with the controller and operates automatically.

The great majority of telphers are built with two bogies, to one of which is attached the travelling motor, which drives all the wheels of that bogie by chain or spur gearing, the other bogie being undriven. Occasionally double bogie telphers are built, having all their travelling wheels driven, thus giving higher acceleration and enabling the machine to climb steeper gradients than can be climbed by a telpher having only one driven bogie. Usually, however, the single bogie drive gives ample travelling power and acceleration. An interesting example of the all-wheel-drive telpher is the large coal-handling plant* at the Nechells Station of the Birmingham Corporation Electricity Department. The track is of the bottom flange type, and each telpher has two four-wheeled bogies, each bogie being driven by two small symmetrically-mounted motors, *i.e.*, each telpher is driven by four small travelling motors.

The travelling and hoisting motors are both under the direct control of the driver in the cabin, either by means of one controller in conjunction with a change-over-switch, or by two independent controllers. In the latter case, simultaneous hoisting and travelling is possible, but in telpher work this is seldom done, the chief reason being that the cross bracings of the supporting trestles form an obstruction. The same obstacle prevents simultaneous travelling and lowering under the brake, but these simultaneous movements

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* By Messrs. The Mitchell Conveyor and Transporter Co., Holborn Viaduct. See article in " Engineering," 23rd Feb., 1923.

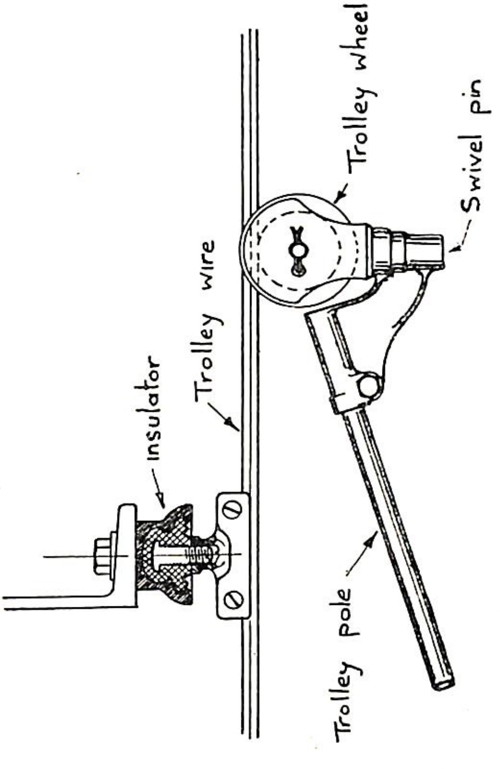
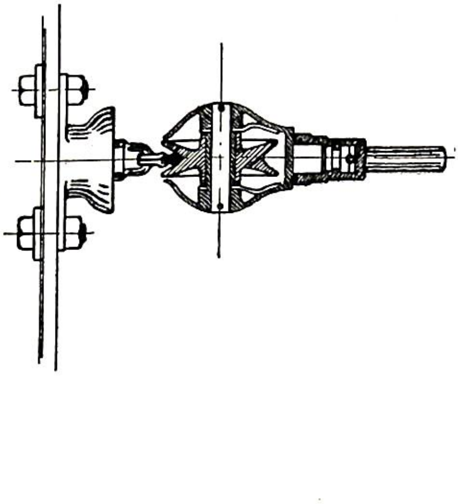

Insulator
Trolley wire
Trolley wheel
Swivel pin
Trolley pole

Fig. 3



can be and are carried out when the supports are of such a form that they offer no obstruction. (See page 36).

Current is collected from bare overhead conductors by means of small swivelling collector wheels mounted on pivoted flexible poles, as on electric tramcars (see Fig. 3).

The horse-power required for the hoisting motor may readily be calculated by the usual methods, but it may be mentioned that a very convenient rough rule is to take one horse-power as being equal to 10 foot tons per minute, this being equivalent to an overall mechanical efficiency of 68%, which is certainly on the low side for a double-spur reduction gear.

Example.—H.P. of motor required to lift 2 tons at 70 feet per minute

$$= \frac{2 \times 70}{10} = 14 \text{ H.P.}$$

Horse-power of Travelling Motor.—The horse-power required for the travelling motor may be approximately calculated as in the following example :—

Let the weight of loaded telpher be 6 tons = 13,440 lbs. Also, let it be required to accelerate the telpher from rest to full speed of 600 feet per minute (=10 feet per second) in a distance of 30 feet.

From the usual relationship, $v^2 = 2fs$.

$$\text{Acceleration, } f = \frac{v^2}{2s}$$

$$= \frac{10^2}{2 \times 30}$$

$$= 1{\cdot}67 \text{ ft. per sec. per sec.}$$

The time required will be, from the relationship $v = ft.$,

$$t = \frac{v}{f}$$

$$= \frac{10}{1{\cdot}67} = 6 \text{ seconds.}$$

The force required will be, from the relationship $P = \frac{Wf}{g}$,

$$P = \frac{13440 \times 1{\cdot}67}{32{\cdot}2} = 700 \text{ lbs. nearly.}$$

$$\therefore \text{Average H.P. required to accelerate} = \frac{\text{ft. lbs. of work done per second}}{550} = \frac{P\,s}{550\,t}$$

$$= \frac{700 \times 30}{550 \times 6}$$

$$= 6{\cdot}36 \text{ horse-power.}$$

Note that this is the **average** H.P. for a constant torque exerted by the motor, and that the horse-power required from the motor at the moment of approaching 10 feet per second is therefore exactly double this, *i.e.*, 12·72 H.P.*

There is an additional resistance to overcome, *i.e.* rolling and bearing friction, which may either be expressed as a certain number of lbs. per ton weight of machine or may be allowed for in the form of overall mechanical efficiency. The latter method will be adopted in this example.

Taking the overall mechanical efficiency at 85% (ball-bearings throughout), the horse-power required will be

$$\text{H.P.} = \frac{12{\cdot}72 \times 100}{85} = 15 \text{ H.P.}$$

There is a still further resistance to be allowed for, and one which is sometimes overlooked, viz., the inertia of the rotating parts, the most important of these, owing to its high speed, being the armature of the motor.

The writer has calculated the moments of inertia of several typical armatures of telpher travelling motors, and on the basis of one revolution of the armature to one foot of travel of the telpher (*i.e.* 600 r.p.m. at 600 feet per minute), he finds that this adds nearly five per cent. to the power required to accelerate.

Adding this percentage to the figure obtained above,

$$15 + \frac{15 \times 5}{100} = 15{\cdot}75, \text{ say } 16 \text{ max. H.P.}$$

But the motor is only required to exert this torque for a total time of 12 seconds in each duty cycle, and as the power required to merely maintain full speed is quite small, the load factor is one which fully justifies doubly overloading the motor during acceleration.

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* There is a more direct way of calculating the H.P. required to accelerate, as it does not require the value of t and avoids conception of average horse-power.

As before, $f = 1{\cdot}67.$
and $P = 700.$

We require to know the H.P. being exerted at the rail when the tractive force $P = 700$ and the velocity is 600 feet per minute. Then, clearly,

$$\text{Maximum H.P.} = \frac{700 \times 600}{33{,}000} = 12{\cdot}72$$

TRAVELLING MOTORS

Diagram gives average Horse Power (per ton weight of telpher) required for acceleration, allowing 85% mechanical efficiency, & 5% for revolving parts

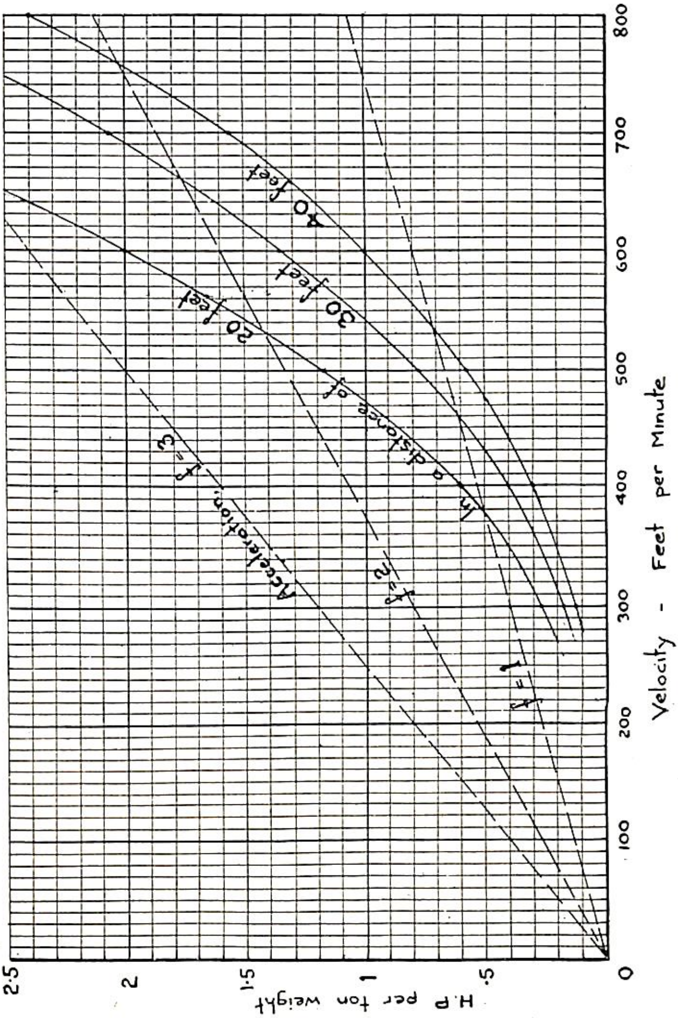

Fig. 4

For normal telpher duty these values may be used for the actual H.P. of the travelling motor (per ton weight of loaded telpher) when the motor is one hour rated

We therefore reach the final figure of

8 H.P.

If the tractive resistance is 20 lbs. per ton (which corresponds in the above case to 85% efficiency), the horse-power required to maintain a speed of 600 feet per minute will be

$$\frac{6 \times 20 \times 600}{33000} = 2{\cdot}2 \text{ H.P. nearly.}$$

To avoid the labour of calculating the horse-power of the travelling motor for every case, Fig. 4 may be used. This diagram gives the H.P. of motor required *per ton weight* of loaded telpher for all cases of velocity or acceleration likely to be required.

The values given allow for an overall mechanical travelling efficiency of 85%, plus 5% for the inertia of the revolving parts. The horse-power may be selected from the curved lines if velocity in feet per minute in a given distance is required, or from the straight lines radiating from the origin if the acceleration is required in terms of feet per sec.^{2}.

Example 1.—Required the H.P. of travelling motor for a loaded telpher weighing 7 tons, to accelerate to a speed of 500 feet per minute in a distance of 30 feet.

From the diagram, the H.P. per ton is ·8.

$$\cdot 8 \times 7 = 5{\cdot}6 \text{ H.P.}$$

Example 2.—The above telpher is required to accelerate at 2 feet per sec.^{2}.

From the diagram, the H.P. per ton is 1·35,

$$1{\cdot}35 \times 7 = 9{\cdot}5 \text{ H.P.}$$

Angle of Bogie Swivel.—When designing the telpher it is necessary to know the maximum angle through which the bogies will turn in relation to the telpher frame when the machine is on a curve, as shown in Fig. 5. It is, of course, an easy matter to calculate or find graphically this angle, but it is still easier to use the following approximate formula :—

$$\text{Angle in degrees} = \frac{\text{Wheelbase}}{\text{Radius}} \times 29$$

The derivation is as follows—

By geometry, the two angles marked θ in Fig. 5 are equal.

$$\text{Sin } \theta = \frac{W}{2} \div R$$

For small values of θ, sin $\theta = \theta$ (Radians)

$$\text{hence } \theta \text{ (radians)} = \frac{W}{2R}$$

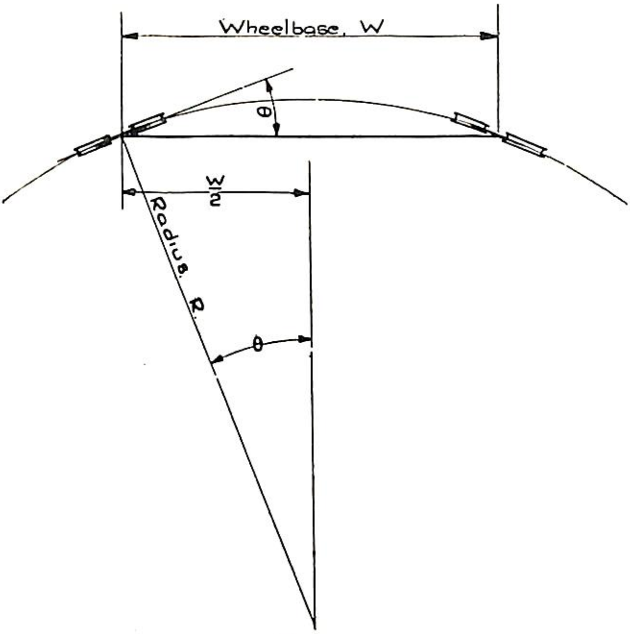

Fig. 5

But a radian = 57·3 degrees.

$$\text{Therefore } \theta \text{ (in degrees)} = \frac{57{\cdot}3\,W}{2\,R} = \frac{28{\cdot}7\,W}{R}$$

$$\text{This may be approximated to } \theta = \frac{29\,W}{R}$$

or even to $\theta = \frac{30\,W}{R}$, an easily remembered formula.

The angle may be obtained even more rapidly from the diagram (Fig. 6).

BALANCE OF THE TELPHER MACHINE.

The distribution of masses in the general layout of the telpher is important, since not only must the weight be suitably shared by the two bogies, but the whole machine must be balanced about a vertical plane through the bogie wheels. A machine would certainly run, even if badly out of balance, but in addition to being very unsightly, the wheel flanges on one side would quickly wear.

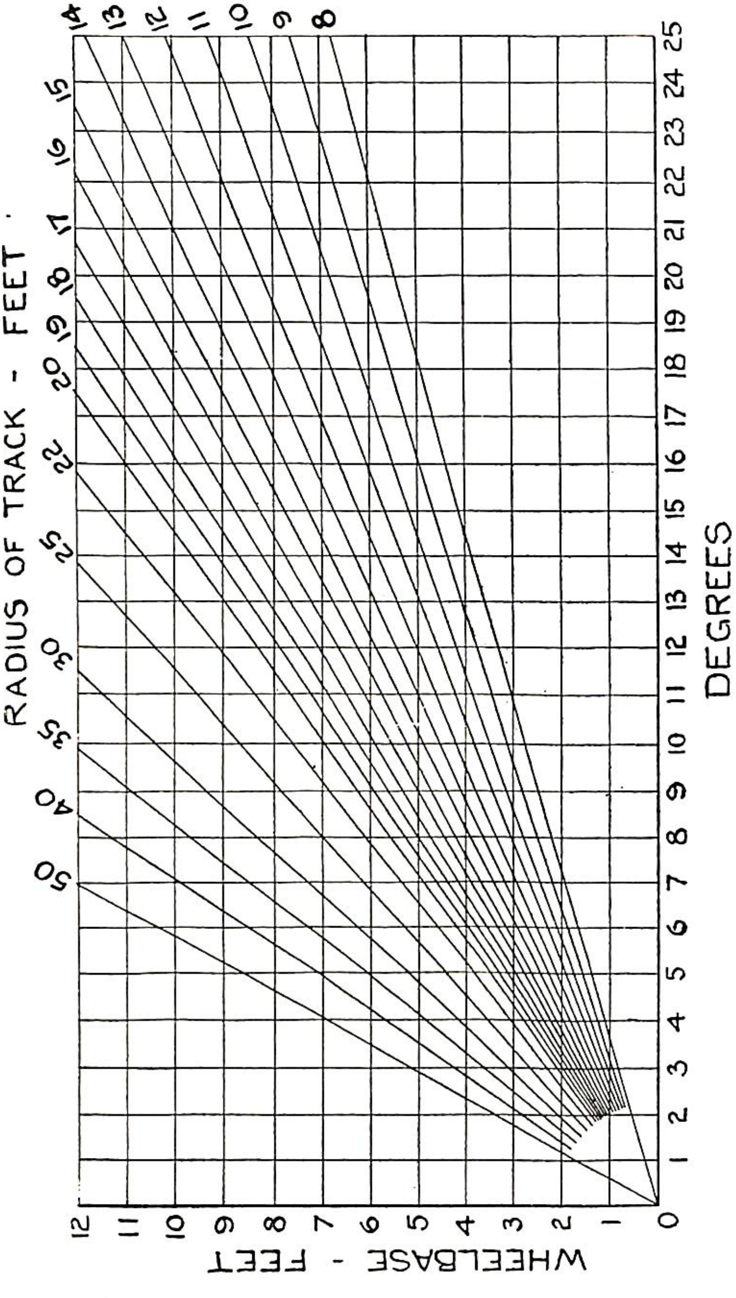
RADIUS OF TRACK - FEET
14
13
12
11
10
9
8
15
16
17
18
19
20
22
25
30
35
40
50
DEGREES
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
WHEELBASE - FEET
0
1
2
3
4
5
6
7
8
9
10
11
12

Fig. 6

The balance must always be calculated for a new design or layout, and this is a simple but laborious process, as the weights and approximate centres of gravity of all parts must be known or calculated, and their moments about the centre line obtained. The various adjustable masses, such as hoisting motor, gearing, and electrical starting resistance, may be so disposed as to give any desired degree of calculated accuracy of balance, but in practice, for an average size of telpher, an unbalanced moment of 1000 inch lbs. does not produce a noticeable inclination.

If the trouble be taken to calculate the position vertically of the centre of gravity, the inclination of the machine from the vertical for any unbalanced moment can be found, as shown in Fig. 7.

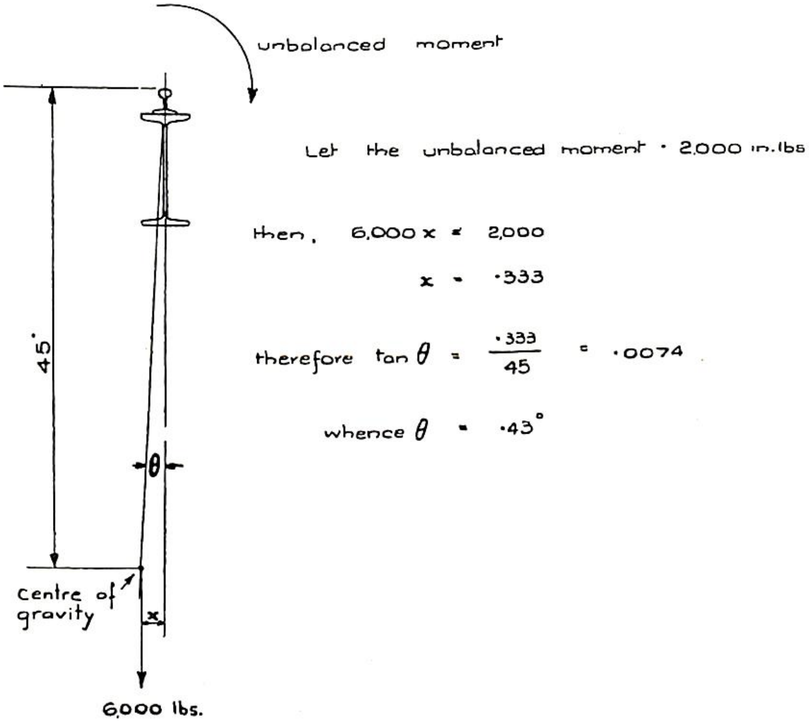

Fig. 7

The method is an approximation, as it requires cos θ to be equal to 1, and it neglects the effect which the distance x has on the arm of the unbalanced moment, but these errors, for the small angle involved, are well within the limits of accuracy of the rest of the data available and the result required.

A more accurate balance is necessary for machines which run on the bottom flange of a rolled steel joist, for the double track gives lateral stability to the bogie and flexible connections must be provided between the bogie and the body of the machine. The centre of gravity of the machine is therefore vertically much nearer the pivots, also the weight involved is rather less (by the amount of the bogies, which being above the pivots do not enter into the balance).

THE TWO SYSTEMS—TOP RAIL AND BOTTOM FLANGE.

1. **The Top Rail System.**—A single running rail is fastened to the top flange of a rolled steel joist, which is fastened on one side of its web to suitable supports. The bogies, which each have two wheels in tandem, are flexibly connected to the body of the machine by stiff hook-shaped links. The principle of this method will be clear from Fig. 2.
2. **The Bottom Flange System.**—The track consists of a rolled steel joist (or channels back-to-back) connected by its top flange to overhead supports. The bogies, which are four-wheeled (or side-guided two-wheeled), run on the two inclined upper faces of the bottom flange.

The second system will be more familiar to the general engineering reader, as it has been in use for a great number of years for hand runways and electric travelling pulley blocks, but the top rail system has been much more extensively used for heavy man-driven telphers, and although in general there is not much to choose between the two systems, it is probable that the top rail system has a higher overall mechanical efficiency in travelling, due to the smaller number of gears and bearings, and to the absence of tread grinding when rounding the curves. The bottom flange carriage really runs on two tracks, spaced about 3½-in. to 4-in. apart, and tread grinding necessarily occurs on the curves in the absence of a differential drive, which for such a case is not justified.

The difference in the length of the two curved paths for a given gauge is quite independent of the radius of the curve and depends only upon the angle turned through, *e.g.* if the paths are 4″ apart, the outer wheel travels 6″ farther than the inner wheel in making a right angle turn.

The bottom flange system is, however, a little cheaper in first cost, and is also very useful where headroom is limited, the amount gained when compared with the top rail system being at least two feet.

An argument sometimes used against the bottom flange system is that the runway joist itself is gradually worn away by the wheels, but in practice it is surprising how many years the joist will last.

Renewable running strips are sometimes screwed or bolted on, but when this is done they should be made fairly thick and the fixing screws or bolts should not be too widely pitched, or the running strip, which has a tendency to "roll out," will buckle and cause uneven running.

Wheel slip is a bigger factor than load in its effect on the life of rails and wheels, and for this reason a too-powerful travelling motor and over-fierce brakes are things to avoid.

The main factors affecting the design of the structure are in general the same for both systems, the chief differences being in the design of the bogies and the methods of supporting the runway joist.

BENDING MOMENTS IN THE RUNWAY BEAM.

The maximum bending moments in independent lengths of runway beam are obtained in the usual manner for rolling loads, and since the pairs of wheels in each bogie are close together compared with the wheelbase, it is assumed that the load is applied to the rail at the centre of the bogie, the error involved being negligible. If the bogie loads are equal, the maximum bending moment occurs when one bogie is at a distance from the centre of the beam equal to one quarter of the wheelbase, as illustrated in Fig. 8, but if the span is less than 1·707 times the wheelbase,* the maximum bending moment occurs when one bogie stands on the centre of the beam.

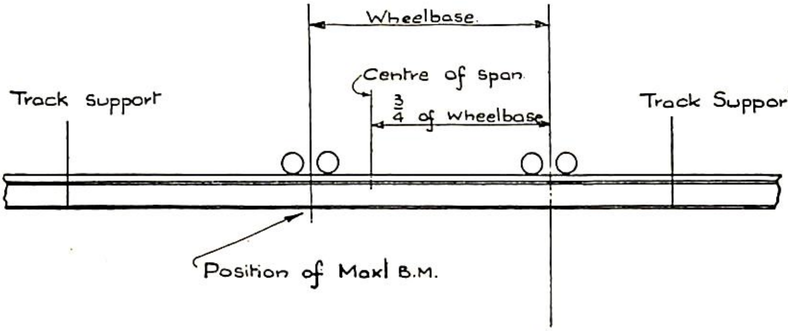

Fig. 8

If the bogie loads are not equal, we may make use of the following theorem, the proof of which may be found in Morley's "Theory of Structures":—

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* Some handbooks state "twice the wheelbase," but it is not difficult to show that this is incorrect. The exact figure is $1 + \frac{\sqrt{2}}{2}$

> " When a series of wheel loads pass over a beam simply supported at its ends, the maximum bending moment under any given wheel occurs when its axis and the centre of gravity of the whole load on the span are equidistant from the centre of the span."

Obviously, only the bending moment under the more heavily loaded bogie need be found.

It is usual to treat each section of beam between two supports as a simple beam, whether jointed at the adjacent supports or not, as the use of the continuous beam theory is not justified by the general conditions of support on a relatively flexible structure. It is not permissible to include the running rail as assisting the strength of the joist, as it is only fastened with widely-spaced clips or bolts to facilitate renewal.

The lateral stability of the beam must also be considered ; it is necessary to limit the ratio of span to flange breadth in order that the beam shall not be unduly flexible or weak in a horizontal direction, this being more necessary with the top rail system than with the bottom flange system owing to the load being applied to the top flange, *i.e.*, above the level of the supports. Some authorities recommend, from experiment, that when the laterally unsupported length of beam exceeds 30 times the compression flange breadth, the maximum safe stress should be reduced, in proportions which will not be quoted here as they are not applicable to telpher beams. Experience has shown that for top rail telphers the span may be made 40 times the flange breadth without additional lateral support being necessary. For an R.S.J. with a 6″ flange, this gives 20 feet as a maximum span, but for spans above this the writer is of the opinion that lateral support should be provided, either by angular struts from the trestles, or by the addition of a channel riveted (flanges downwards) to the top flange of the joist, the rail being then fastened to the channel. It must be pointed out that this latter method is somewhat extravagant, for the addition of extra metal to one flange only of a beam is of almost negligible assistance to the beam as regards vertical strength. The strength could be very considerably increased by adding metal to the bottom flange also, say by rivetting a plate on, but the obtaining of larger spans by this means is of doubtful economy, though a still greater objection is the vertical deflection of the beam which results from a high ratio of span to depth, causing a wrenching action at the supports.

The maximum fibre stress in top rail runway beams should not exceed 6 tons per square inch, and in bottom flange runway beams a still lower figure should be used owing to the secondary stress produced by the wheels tending to bend the bottom flanges locally.

A serious difficulty arises at the curved portions of the track in deciding how frequently the runway joists shall be supported, for the sections of joist between supports are curved in plan, and this is a proposition in the strength of materials which the writer admits is outside his ability to deal with, as it involves, among other considerations, the resistance of irregular sections to torsion, which in itself is a highly complicated study. Those interested in the general problem are referred to a book by Gibson & Ritchie, entitled "The Circular Arc Bow Girder" (Constable), and "The Stress Analysis of Bow Girders," by Professor Sutton Pippard and F. T. Barrow (H.M.S. Office, 1/3).

A purely empirical method, which the writer makes no attempt to justify, but which has in practice proved satisfactory for 14″ × 6″ beams, is to limit the torsion arm, or chordal overhang, to about 5″. It will be seen that this tends to shorten or lengthen the

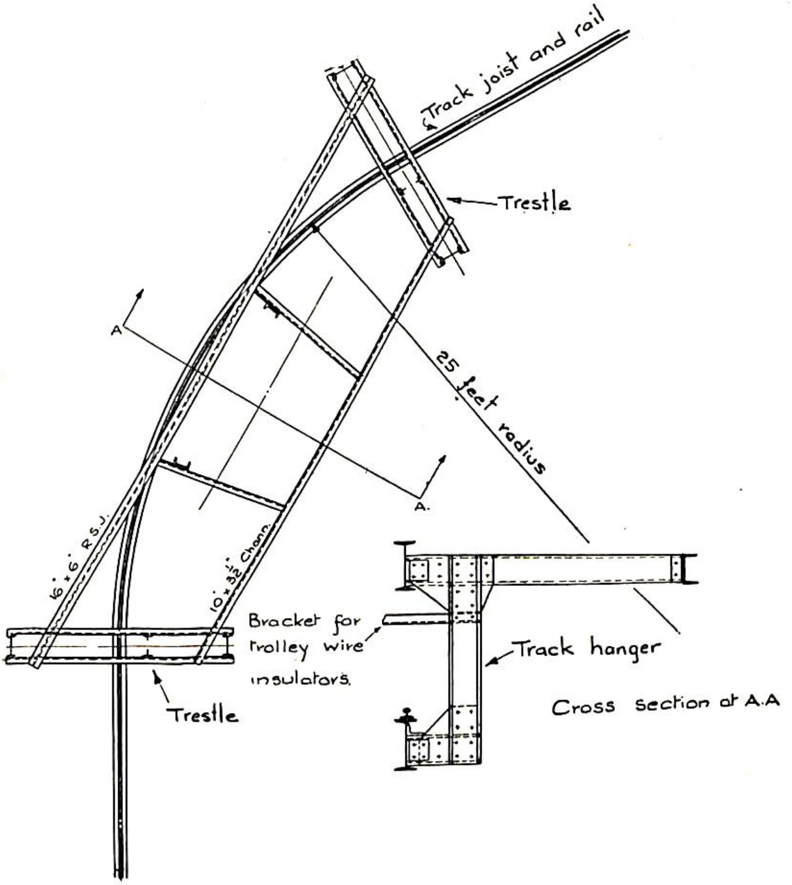


Fig. 9

distance between supports according to the radius of the curve, and is thus quite rational, but modification is sometimes necessary due to other considerations. Fig. 9 shows a typical method of supporting the runway joist at a curve. It will be noticed that a second beam is used, to which the top cross channels of the track hangers are attached, in order to provide lateral stiffness.

CONNECTION OF TRACK JOISTS TO TRESTLES.

Top Rail System.—Tracks of height up to about 35 feet are usually supported by a series of trestles, spaced apart at a distance which can safely be spanned by the runway joist, and this distance varies according to the weight of the machine and the section of joist used, but is usually about 20 to 24 feet. The top portion of such a trestle is shown in Fig. 10, which also shows two common methods of attaching the runway joist to the trestle top beam.

In both cases the maximum bending moment in the top beam is greater than if the load were slung from the centre of the beam. In the following calculations the knee struts are neglected, as being incapable of preventing deflection, and the top beam is taken as simply supported at the ends.

Case 1.—The track joist is attached to an arm connected to the trestle leg and hung by a suspension member attached off-centre to the top beam. It is, of course, not a difficult matter to calculate the maximum bending moment from first principles in every case, but there is no need for this, as the following proof shows that the lighter reaction is always $\frac{W}{2}$, whence the maximum bending moment is in every case obtainable by a single multiplication.

Referring to Fig. 10, Case 1, the pull on the suspension member is

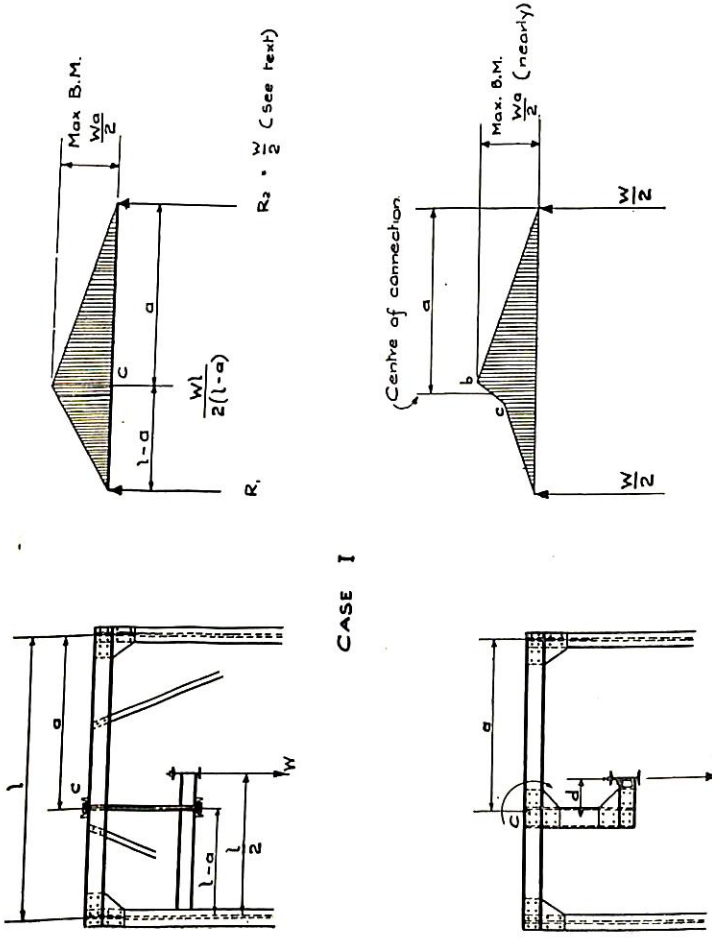
$$W \times \frac{l}{2} \div (l-a) \ i.e., \ \frac{W\,l}{2\,(l-a)}$$

The right-hand reaction $R_2 = \frac{W\,l}{2\,(l-a)} \times \frac{l-a}{l} = \frac{W}{2}$

Hence the bending moment at C is $\frac{W\,a}{2}$

This is a useful and easily remembered fact.

Case 2.—The track joist is suspended from the top beam by means of a rigid "hanger" attached off-centre.



CASE I

CASE 2

Fig. 10

No calculation is here necessary to show that the right-hand reaction is equal to $\frac{W}{2}$, hence we again have the maximum-bending moment equal (very nearly) to $\frac{Wa}{2}$, a being the distance of the centre of the connection from the support. The stresses in the beam are those which would be produced by a load W applied vertically downwards at C, plus a couple of magnitude W*d*. The effect of this couple is shown by the bending moment diagram. Strictly, the line *b c* in the bending moment diagram should be staggered, corresponding to the rivets of the connection.

It will be found that in general the extra bending moment produced by the off-central connection of the hanger or suspension bar is nearly 25 per cent.

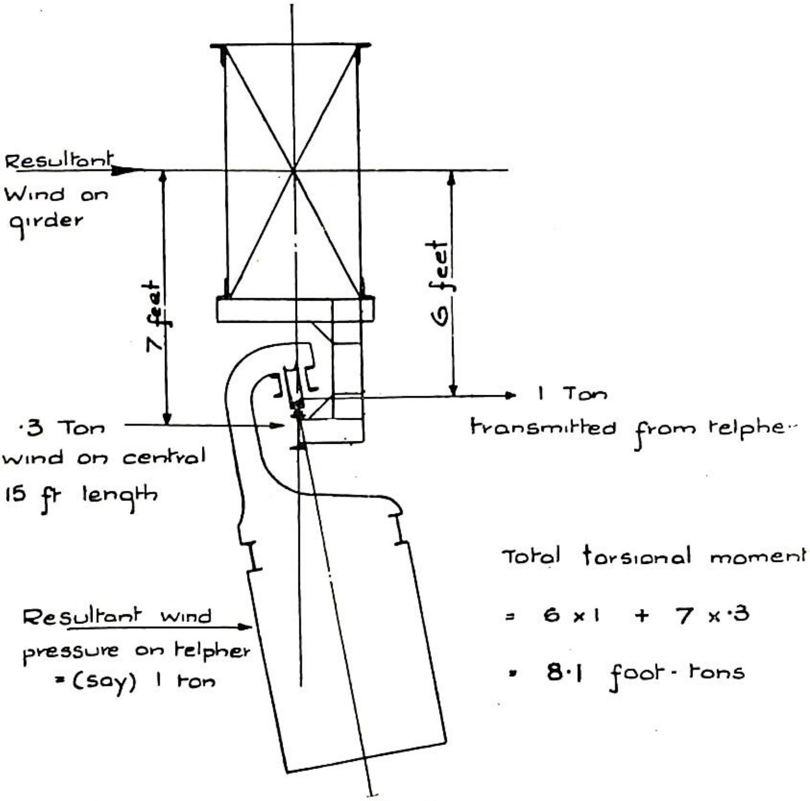


Fig. 11

Bottom Flange System.—If the runway consists of a rolled steel joist, it is usual in telpher work to hold up the joist by means of a group of bolts in tension passing through the top flange. In light hand runways and small power runways such joists are frequently held by specially-shaped clips, which do not require the joist to be drilled, but for heavy loads such clips are not used. It is not sufficient to use the same factor of safety for these tension bolts as that used for the structure, as, owing to such causes as defective track joints, the travelling telpher can apply shock loads to the suspension bolts. The effect of this can be minimised by using bolts of large volume (length being quite as useful as area), not overlooking that the criterion of direct strength is the area at the bottom of threads. Three tons per square inch is not too low a figure for maximum stress in these bolts.

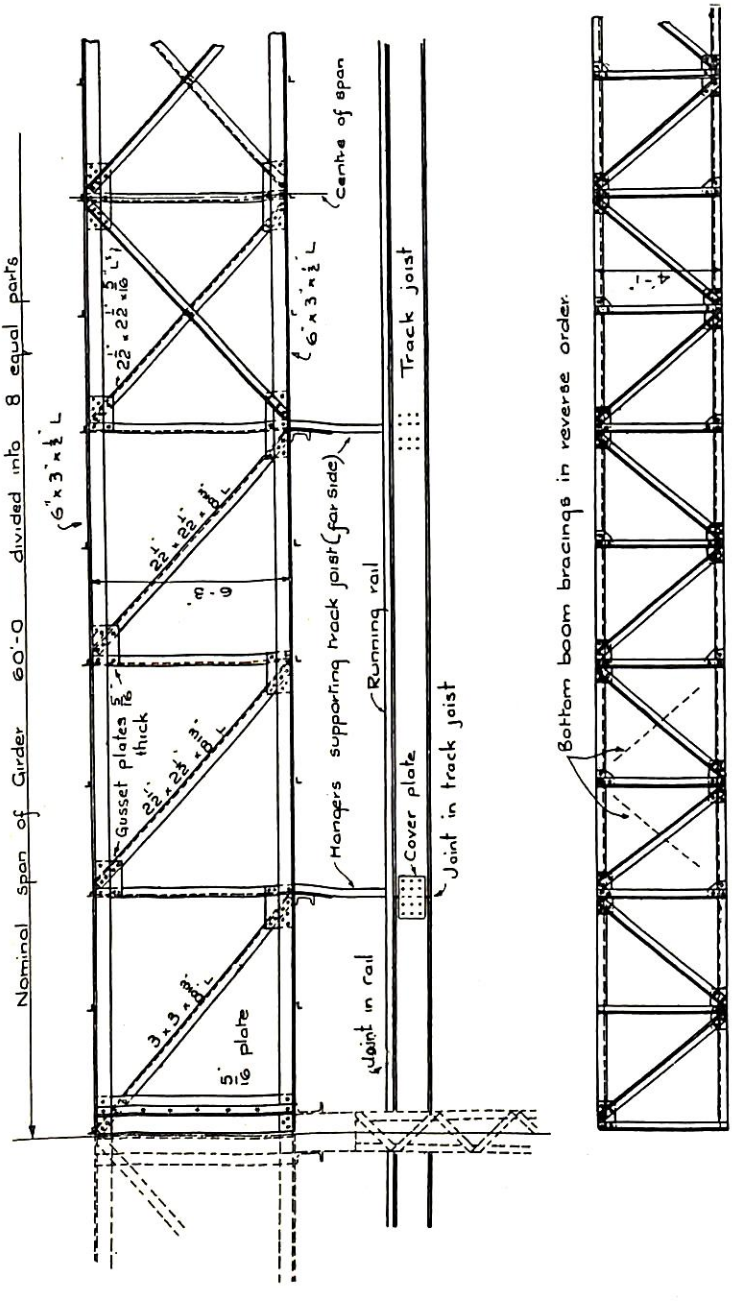
When two channels are used back-to-back, it is possible to make a very good connection by means of a gusset plate between the webs of the channels, fastened by double countersunk rivets in double shear. This leaves a narrow gap between the channels throughout the whole length of the track, and this should be filled if corrosion is to be avoided, for it is almost impossible to paint it.

DESIGN OF A 60-FEET SPAN LATTICED GIRDER.

Auxiliary girders for supporting the runway joist must be employed in cases where the distance between trestles is too great to be spanned by the runway joist unaided. For spans which are not much outside the capacity of the joist itself an auxiliary rolled steel joist may be used to bridge the tops of the trestles, the runway joist being supported from this at one or more places by a hanger built up from mild steel sections, similar to those shown in Fig. 9, but usually, if additional supporting girders are required at all, it is more economical to further increase the span and support the track joist from a latticed box girder. Many forms of construction have been used by different designers for this purpose, and sometimes apparently small considerations greatly affect the design. One much-used form of construction is illustrated in Fig. 12, which shows one-half of a symmetrical 60-feet span girder, and the methods employed in the design of this girder will now be described.

Before giving calculations it will be well to explain the assumptions made.

General Assumptions.—(1) Vertical loading. The girder is treated as being simply supported at its ends. For general telpher girders the continuous beam theory cannot be considered, especially as a slight but uneven subsidence of trestle foundations is quite usual. Having made this assumption, any attempt to connect the



PLAN OF TOP BOOM.

Fig. 12

girder top booms at the trestles will upset the basis of the calculations, and the bottom booms only are connected by a substantial bearing plate on the trestle top, and by a short inside connecting angle which also increases the bearing area.

By experience it is known beforehand that a girder of this span, for a 6-ton telpher, will weigh about $3\frac{1}{2}$ tons, and for convenience this is divided into seven equal parts and applied at the top panel points, the errors being on the right side.

The track joist is attached to the girder at equal intervals of 15 feet, and is not in this case attached at the trestles.

The nett weight of the telpher machine is taken at 4 tons, and it carries a gross load of two tons. It is usual to take the equivalent static rolling load as equal to the nett weight of the telpher machine, plus twice the gross load lifted, in order to make allowance for sudden lifting or braking of the load, and this brings the total equivalent rolling load to 8 tons in the case illustrated. There is no particular justification for exactly doubling the load, as greater forces in the lifting ropes can be produced by careless snatching at the load or by fierce braking during lowering.

The track joist, including rail, fishplates, etc., is taken at 70 lbs. per foot, and the track hangers at ·12 ton each. The fact that the weight of the track hangers is not centrally applied (laterally) may be ignored.

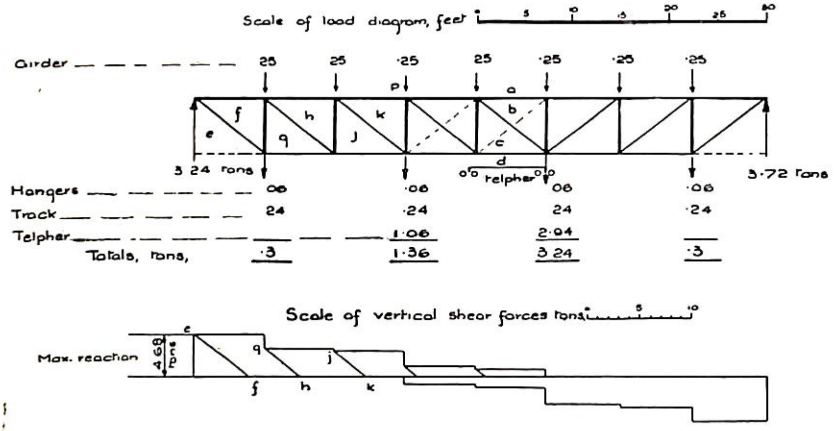
(2) Wind loading. Wind pressure is taken at 30 lbs. per square foot for directly exposed surface, and at 15 lbs. per square foot for partially-masked surface, *e.g.*, the leeward vertical bracing.

Allowance for the effect of wind on the telpher machine and the track joist presents a very difficult problem. It is quite usual to divide the wind load on these equally between the top and bottom wind bracings, but reference to Fig. 11 will show that, in addition to a bending moment, the wind on the telpher machine and on the mid-span of runway joist applies to the girder a twisting moment of over 8 foot tons.

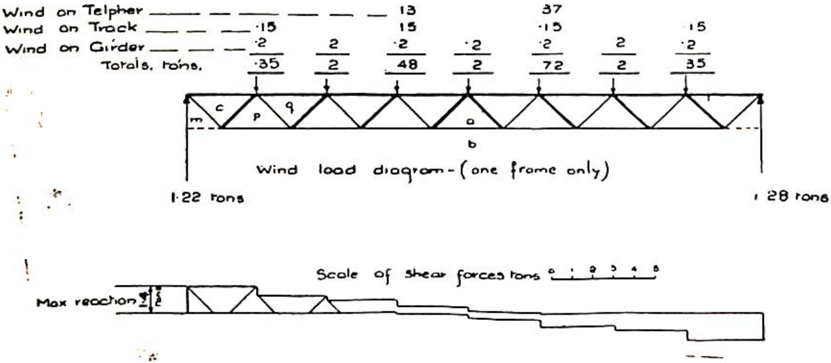
Now it is quite possible, by using Castigliano's method of least work, to calculate the additional stresses due to this twisting moment, but the work is so complicated and laborious, and the results so dependent upon constructional accuracy, that its use is not justified in such a case.

This is certain, that the consideration of the load as eccentrically applied, compared with central application, alters the stress in all diagonal bracings, whether horizontal or vertical, in a manner which is not statically determinate, and it would appear to increase the stress in the bottom wind bracing and in the windward vertical bracing, and to relieve the stress in the topside and leeward bracings. The writer's opinion is that the effect on the vertical bracing may safely be neglected, and that the bottom wind bracing should be designed to carry a major portion of the total wind load, but in the subsequent calculations the usual practice is followed and the load is shown equally divided.

Calculations.—From the foregoing notes the frame diagrams on Fig. 13 will be readily understood. In both cases, *i.e.*, for vertical and side loading, the loads are given for one frame only, hence the end reactions are half the total, and this must be remembered when designing the trestles.



Combined Shearing force diagram for rolling and dead load (one frame only)


Wind on Telpher 13 37
Wind on Track ·15 15 ·15 ·15
Wind on Girder ·2 2 ·2 ·2 ·2 2 ·2
Totals, tons, ·35 2 48 2 72 2 35
c g m p a b
Wind load diagram - (one frame only)
1·22 tons
1·28 tons
Scale of shear forces tons 0 1 2 3 4 5
Max reaction

Combined shearing force diagram for steady wind load on girder and rolling wind load on telpher

Fig. 13

For a girder of 60 feet span it is scarcely economical to use different boom sections corresponding to the load, and if the same section is carried right through, a rolling load diagram for bending moments is unnecessary, as the maximum stress will occur in the central boom members when one telpher bogie is on either of the track hangers nearest the centre of the girder (*not* when the telpher is central on the girder. The difference is appreciable; in our example the load in *a b* for the position of telpher shown is 12·2 tons, but with the telpher central the load is only 11·4 tons). This one position of the telpher is sufficient for the purpose of obtaining maximum loads in top and bottom booms, and by the method of moments these are (calculations of reactions not given) :—

Taking right-hand moments about the lower end connection of *b c*.

$$\text{Load in } a\,b = \frac{3\cdot72 \times 22\cdot5 - (\cdot5 \times 11\cdot25 + \cdot3 \times 15)}{\text{Depth of Girder}}$$

$$= \frac{73\cdot5}{6} = 12\cdot2 \text{ tons.}$$

$$\text{Load in } c\,d = \frac{3\cdot72 \times 30 - (3\cdot24 \times 7\cdot5 + \cdot3 \times 22\cdot5 + \cdot75 \times 15)}{6}$$

$$= \frac{69\cdot2}{6} = 11\cdot5 \text{ tons.}$$

The wind loads are :—

	Tons.
Wind on girder (equivalent* depth 2·3 feet) $= \frac{60 \times 2\cdot3 \times 30}{2240}$	= 1·86
Wind on leeward side, 50 per cent. of above,	·93
Total wind on girder,	= 2·8
÷ 2 for one frame	= 1·4
÷ 7 for each panel†	= ·2

---

* The "equivalent depth" of the girder is the depth of an unbroken surface having an area equal to that of the members of the girder, including the top and bottom wind bracing and all gusset plates, *i.e.* the areas are "massed up" vertically to give the equivalent solid depth, as is sometimes done horizontally when finding the section modulus figures of hollow sections. The massing up is only done approximately, and makes the calculation of wind resistance a much simpler matter than would be the case if the wind on various bars were considered separately.

† It would be more nearly correct to divide the weight of the girder into *eight* equal portions, corresponding to the number of panels, but the figure in this case is so conveniently divided by 7 and the error involved is so slight that the approximation is not objectionable.

Wind on track (depth, say $1\frac{1}{2}$ feet) $= \dfrac{60 \times 1\frac{1}{2} \times 30}{2240} = 1{\cdot}2$

Wind on telpher = 1 ton.

The maximum boom loads due to wind pressure are found in a similar manner to that described for vertical loading, the telpher being again displaced from the centre of the girder, so that one bogie is opposite a hanger. Using the method of sections and taking moments about the central connection,

$$\text{Load in } a\,b = \frac{1{\cdot}28 \times 30 - ({\cdot}72 \times 7{\cdot}5 + {\cdot}2 \times 15 + {\cdot}35 \times 22{\cdot}5)}{\text{Width of Girder}}$$

$$= \frac{22{\cdot}1}{3{\cdot}75} = \mathbf{5{\cdot}9\ tons.}$$

Note.—The expressions "depth of girder" and "width of girder," used in the above calculations, refer to the effective depth and width respectively, *i.e.*, the distance apart of the centroids, or neutral axes, of the boom angles, and not the overall depth and width.

Combining the boom loads due to vertical and horizontal loading,

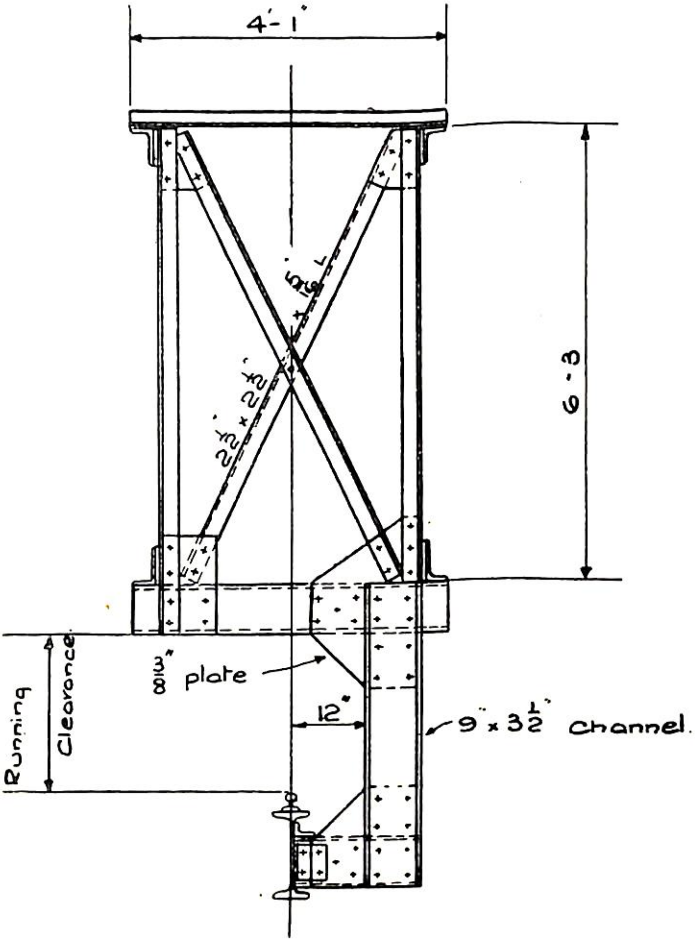
Maximum top boom compression = 12·2 + 5·9 = **18·1 tons.**
Maximum bottom boom tension = 11·5 + 5·9 = **17·4 tons.**

Unequal angles are selected for the top compression members, since the system of bracing gives lateral support twice as frequently as it gives vertical support. For the completely free length between the wind bracing connections the least radius of gyration of the angle must be taken, but for the length between vertical bracing connections the radius of gyration about the neutral axis parallel to the shorter flange may be taken, since on this length the strut can only fail vertically. The size of the angle will be chosen on the greater of the two values for the ratio of slenderness. It is usual to regard these continuous boom members as struts having fixed ends. Whether this is justifiable is open to doubt, but this is not the place for the discussion.

For the maximum load in diagonals and vertical bracings for any position of the telpher machine, combined shearing force diagrams are drawn for the distributed and rolling loads, and the maximum load in any bar is found in the usual manner by drawing across the shear diagram a line parallel to the bracing bar. The shear diagram for vertical loading also shows how far counterbracing should be carried to avoid reversal of stress.

The sizes of bracing members chosen are shown on the drawing of the girder, Fig. 12, and it will be noticed that the sizes, especially in the middle of the girder, are greater than are strictly necessary, this being due to their comparative length. Gusset plates are used throughout for the wind bracing to increase the general rigidity, and because all the wind bracing bars act alternately as struts, and are therefore better double riveted.

Internal cross bracing, or diaphragm bracing, is necessary at intervals to preserve the rectangular form of the girder, and this is shown in Fig. 14, which also shows a track hanger support.



CROSS SECTION THROUGH GIRDER

Fig. 14

The stress in the vertical channel of the track hanger must be considered. The maximum load which can be transmitted to any one hanger on this girder is

	tons.
15 feet of track, with rail, etc.,	·48
$\frac{2}{3}$ weight of telpher (due to wheelbase),	5·33
	5·81

Assuming the hanger channel to be 9″ across the flanges, and the running clearance from the centre line of the rail to be 12″, the load is applied with 16½″ of eccentricity, which gives a combined bending and direct stress.

The bending moment is

$$5{\cdot}81 \times 16{\cdot}5 = 96 \text{ inch tons.}$$

Taking a *bending* stress of 5·5 tons per square inch, the section modulus required is

$$Z = \frac{M}{f} = \frac{96}{5{\cdot}5} = 17{\cdot}5 \text{ nearly.}$$

A $9'' \times 3\frac{1}{2}'' \times 22{\cdot}27$ lbs. channel has $Z = 17{\cdot}7$.

The direct stress must be added, and this is

$$\frac{\text{load}}{\text{cross sectional area}} = \frac{5{\cdot}81}{6{\cdot}55} = {\cdot}89 \text{ tons per sq. inch.}$$

The total stress is therefore $5{\cdot}5 + {\cdot}89 = 6{\cdot}39$ tons per square inch, which is slightly high for this duty.

The suitability of the groups of rivets at the top and bottom corner connections must be verified in the usual manner for eccentric loading by finding the centroid of the group and its moment of inertia.

An alternative end construction for a girder is shown in Fig. 15, and this form is also much used, especially in connection with the Warren type of bracing, but it may be noticed that it is much more difficult to stay the top of the girder to the tops of the trestle legs, and if this is not done additional bending moments due to wind are induced in the trestle top beam.

DESIGN OF AN ORDINARY TRESTLE.

The design of an ordinary trestle is a fairly straightforward matter, but very frequently trestles of special design are required to meet the conditions in their immediate vicinity, and sometimes no two trestles are alike throughout a complete installation. However, even the ordinary trestle, as usually built, presents problems

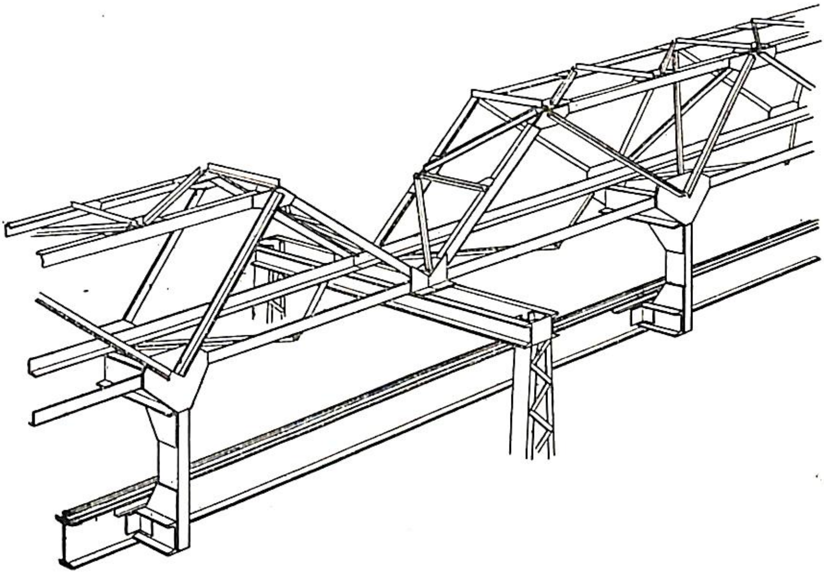


Fig. 15

in design which are by no means easy of exact solution, and the general factors which influence the design of such a trestle will now be considered.

Fig. 16 is a drawing of a trestle designed to carry telpher girders of 60 feet span, such as that described in the last section, which in turn support a runway joist at a height of 40 feet.

The diagram (Fig. 17) shows the loads which the trestle must be capable of resisting. The load due to wind on the latticed girders will be equal to half the total wind load on the two girders carried by the trestle, *i.e.* equal to the total wind load on one girder, if the spans are equal. This load, from the previous section, is 2·8 tons.

The wind on 60 feet of runway is 1·2 tons, as already used for the design of the girder.

The actual wind load on the telpher machine is one ton, but owing to the wheelbase (assumed 10 feet) and the position of hangers on this girder (see Fig. 12) only four-fifths of this load can be transmitted to any one trestle, *i.e.*, ·8 ton. It will also be noticed that, although the centre of pressure of the wind on the telpher machine is several feet below the running rail, the load is shown applied to the trestle at rail level. This is due to the telpher being virtually pivotted on its running rail, which is therefore the only point at which the horizontal load can be resisted. This is of some importance when estimating overturning moments.

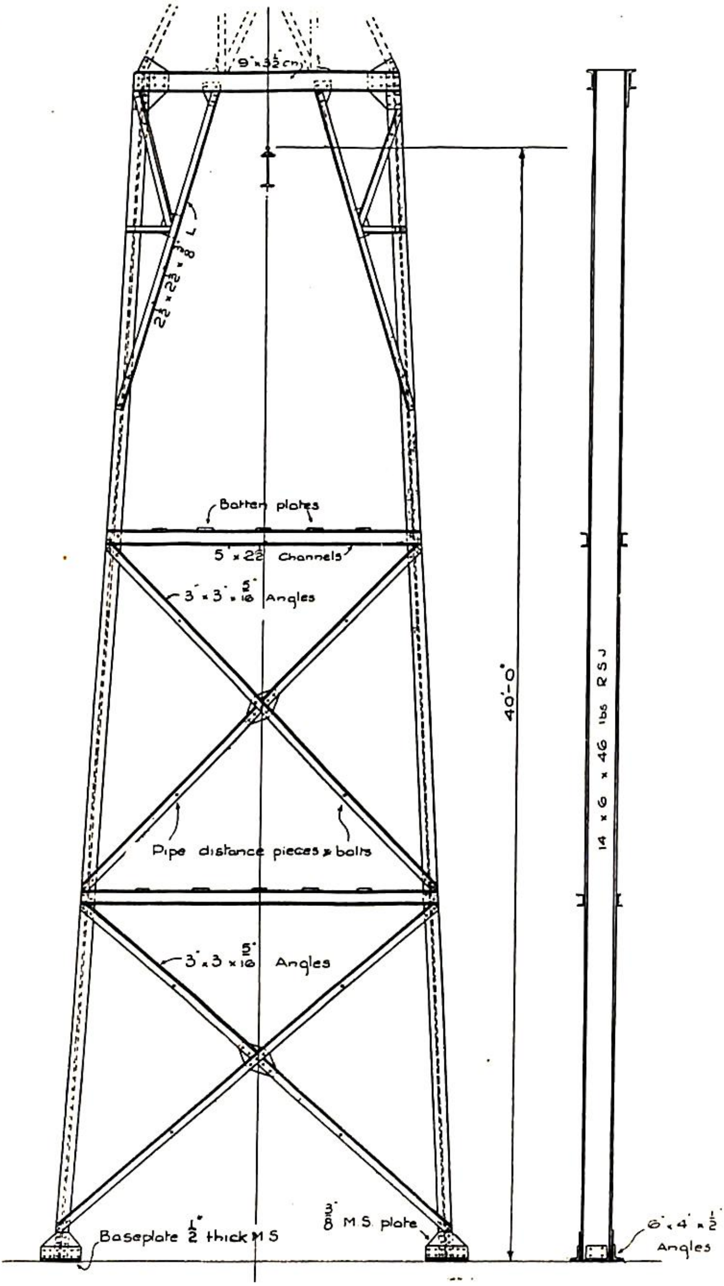

9" × 3½" Cn
2½" × 2½" × 3/8" L
Batten plates
5" × 2½" Channels
3" × 3" × 5/16" Angles
Pipe distance pieces & bolts
3" × 3 × 5/16" Angles
Baseplate ½" thick M S
3/8" M S plate
40'-0"
14 × 6 × 46 lbs R S J
6" × 4" × ½" Angles

Fig. 16

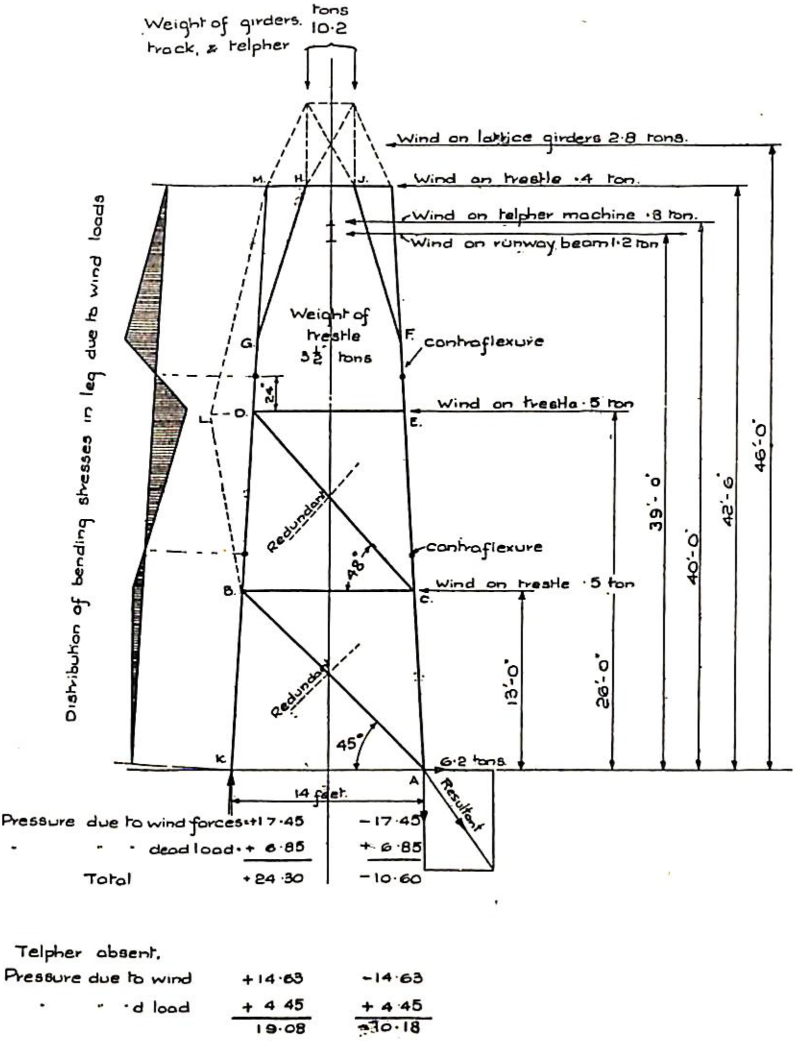

Weight of girders, track, & telpher
tons 10·2
Wind on lattice girders 2·8 tons.
Wind on trestle ·4 ton.
Wind on telpher machine ·8 ton.
Wind on runway beam 1·2 ton
M.
H.
J.
Weight of trestle 3½ tons
G.
F. contraflexure
24"
L
D.
E.
Wind on trestle ·5 ton
Redundant
contraflexure
48°
B.
C.
Wind on trestle ·5 ton
Redundant
45°
K
A
6·2 tons
Resultant
14 feet
13'-0"
26'-0"
39'-0"
40'-0"
42'-6"
46'-0"
Distribution of bending stresses in leg due to wind loads
Pressure due to wind forces +17·45 −17·45
" " dead load + 6·85 + 6·85
Total +24·30 −10·60
Telpher absent.
Pressure due to wind +14·63 −14·63
" " d load + 4·45 + 4·45
19·08 −10·18

Fig. 17

Knowing that the trestle legs will consist of rolled steel joists about 15″ wide, the side wind load on the trestle can be estimated. For a slightly oblique wind, taking the full area of both legs and neglecting bracing,

$$\text{Wind resistance of trestle} = \frac{43 \times 1{\cdot}25 \times 2 \times 30}{2240} = 1{\cdot}4 \text{ tons nearly.}$$

This is, for convenience, applied at three points on the windward side of the trestle, as shown in the diagram of loading (Fig. 14).

The dead loads applied to the trestle are :

$$\begin{array}{lcl} \text{60 feet of girder and track} & = & 5{\cdot}4 \text{ tons.} \\ \text{Telpher } (6 \text{ tons} \times \frac{4}{5}), & = & \underline{4{\cdot}8} \\ & & \underline{\underline{10{\cdot}2}} \text{ tons.} \end{array}$$

The telpher is taken at its net weight (loaded) of 6 tons, as the effect of any suddenly-applied load in the lifting ropes is nearly all absorbed in the girder, and is probably scarcely felt at the foundations.

The weight of the trestle is assumed to be 3·5 tons, which is a little high.

The load diagram (Fig. 17), shows all the loads applied, and the magnitude and directions of the reactions. The cross bracings are designed as ties, therefore the windward foundation takes the whole of the horizontal wind load (neglecting slight angularity of the leeward leg).

The foundation loads with telpher present and telpher absent are shown. The greatest downward pressure always occurs when the telpher is present, but very frequently the greatest uplifting force on the windward foundation occurs when the telpher is absent, and both calculations should be made.

Foundation Loads. Telpher present.

Overturning moment =

$$2{\cdot}8 \times 46 + {\cdot}4 \times 42{\cdot}5 + {\cdot}8 \times 40 + 1{\cdot}2 \times 39 + {\cdot}5 \times 26 + {\cdot}5 \times 13$$
$$= 244{\cdot}5 \text{ foot tons.}$$

Therefore, the reaction at the bases due to wind only is :

$$\frac{244{\cdot}5}{\text{distance apart of bases}} = \frac{244{\cdot}5}{14} = 17{\cdot}45 \text{ tons.}$$

Half the sum of the dead loads is added algebraically at each base to give the gross pressures and uplifts.

In the diagram downward forces at the bases are given the + sign and upward forces the − sign.

Cross Bracing.—Neglecting the slight inclination of the legs, and taking the angle of the bracing at 45° and 48° :

$$\text{Tension in A B} = \frac{6{\cdot}2}{\cos 45^\circ} = 8{\cdot}7 \text{ tons.}$$

i.e., 4·35 tons in each bar (double bracing).

$$\text{Tension in C D} = \frac{5{\cdot}7}{\cos 48^\circ} = 8{\cdot}6 \text{ tons.}$$

This calculation only determines the number of rivets required, since the angle ties, being long, must be proportioned accordingly. Two $\frac{3}{4}''$ diameter rivets in each bar will be just sufficient in this case.

Top Beams.—The stresses in these are very complex, and partly depend upon whether G H and F J are designed as struts or ties, but however strong G H and F J may be, they cannot entirely prevent deflection of the top beam, owing to the thrust at their lower ends causing lateral deflection of the main legs, and although the resistance to this deflection does help the struts to support the top beams, the value cannot practically be even approximately determined. It is best to treat them as simple beams supported only at the ends.

Main Legs.—The stresses in the legs are very difficult to determine exactly, owing to the top panel being a knee-braced portal. To find exactly the positions of the points of contraflexure would be very difficult and laborious, also it would give an unnecessarily exact figure. The position of the points of contraflexure in the portal may safely be assumed to be about halfway between the points E F and D G respectively, and the horizontal loads applied at each of these points will be $\frac{5{\cdot}2}{2} = 2{\cdot}6$ tons, that is, taking the knee braces as struts and dividing the load equally, but it is somewhat doubtful whether this latter assumption is fully justified. The general form of the bending moment diagram for the leeward leg is shown on the left of Fig. 17. It will be seen that the bending action continues right to the base of the stanchions, and this fact should be remembered when considering the legs as struts.

It is often economical to reinforce the legs against bending at the positions of maximum bending moment, rather than select the section of the leg to resist the maximum bending moment.

The bending moment in the leg can be avoided entirely by using external bracing, as shown by the dotted lines B L M, and this is sometimes done on Continental structures. Also, if the knee braces are connected at D and E, the bending moments are avoided, but this takes up telpher machine clearance and tends to make a wider and heavier trestle.

In the design illustrated, the total horizontal shear at the points of contraflexure in the top portal is, as already stated, 5·2 tons, giving 2·6 tons applied at each point. As the points of contraflexure are 24″ from the connections D and E, the bending moments in the main legs at these connections will be

$$24'' \times 2{\cdot}6 \text{ tons} = 62{\cdot}5 \text{ inch tons.}$$

requiring section modulus, $Z = \frac{62{\cdot}5}{\text{Safe stress}} = \frac{62{\cdot}5}{7} = 8{\cdot}9$

The 14″ × 6″ R.S. joist used in this case has $Z = 7{\cdot}2$, and is therefore not good enough without reinforcement, especially as there are two other stresses at this point, namely, shearing stress due to the horizontal load, and compressive stress due to dead load and overturning moment. These are easily calculated and added, but they are relatively small.

The portion B K of the leg has the heaviest column load to resist, and this is 24·3 tons.

For × 14″ × 6″ R.S. joist, $\frac{l}{r} = \frac{13 \times 12}{1{\cdot}26} = 124.$

For both ends fixed, and using Claxton Fidler's formula,

Safe load = 48·5 tons.

The leg is required to take 24·3 tons, but the margin is useful in view of the bending stress already referred to, also it is doubtful whether the ends may truly be considered as "fixed."

Trestle Bases.—It is convenient in our typical case to make these about 2′-0″ by 1′-6″, giving a pressure area of 3 square feet and a foundation pressure of

$$\frac{24{\cdot}3}{3} = 8{\cdot}1 \text{ tons per square foot,}$$

which is quite satisfactory for a mass concrete foundation.

Four long bolts, each $1\frac{1}{4}$″ diameter, and held into the concrete by anchor plates, will be sufficient for each base, and the stress in these bolts on the windward side will be

$$\frac{\text{Uplifting force}}{\text{Total area at bottom of threads}} = \frac{10{\cdot}6}{3{\cdot}56} = 3 \text{ tons per sq. in.}$$

DESIGN OF THE OPEN TRESTLE.

Trestles having no cross bracing between the legs are sometimes very desirable, *e.g.* when a long stockpile of coal is to be formed under the line of the track and reclaimed by grab from the telpher. In such a case absence of cross bracing allows of quicker and safer working. Such a trestle presents an interesting problem in design, and although the two braced legs and deep top cross beam give a combination which is easily determinate, the chief difficulty is to obtain sufficient rigidity, and generally such a trestle is rather heavier and of a more expensive construction than the ordinary trestle. Nevertheless, its adoption is sometimes fully justified, and as it is likely to be increasingly used in connection with telpher plants, a detailed description is here given of one method of designing such a trestle.

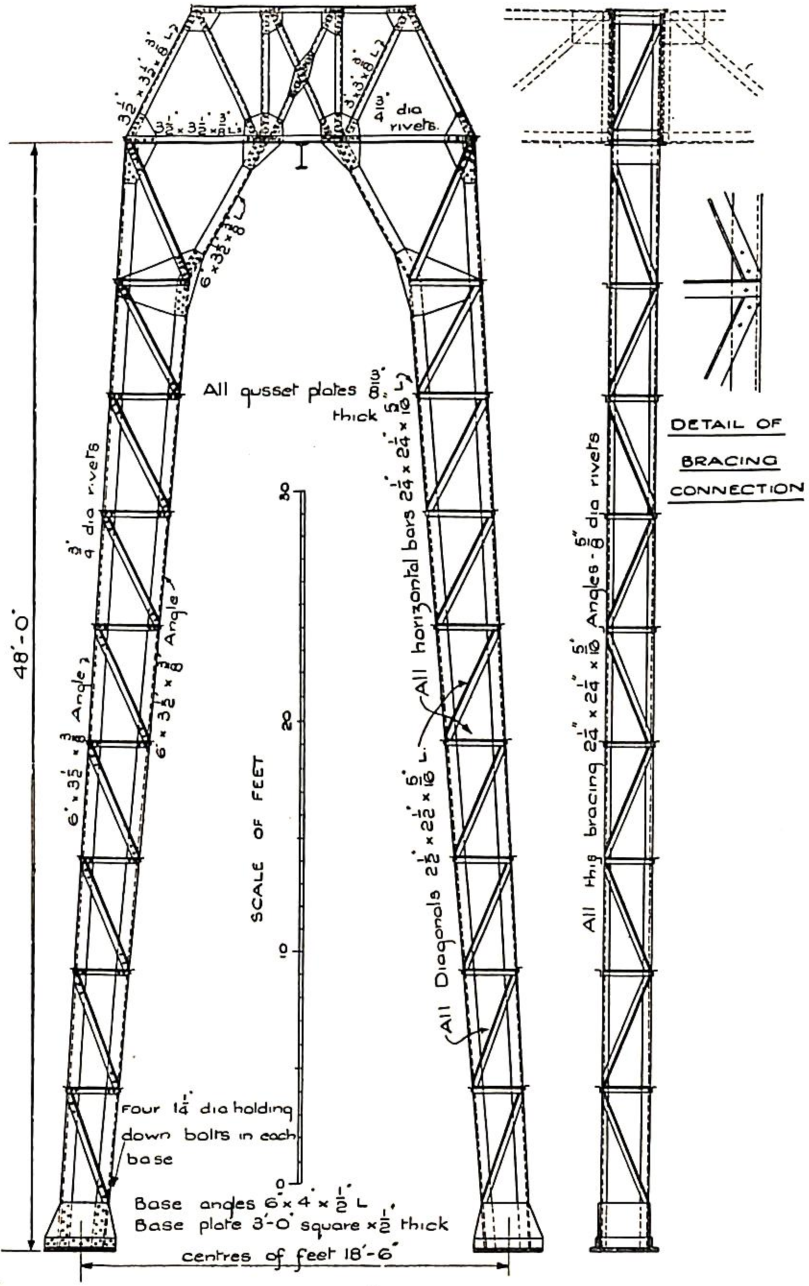

3½" × 3½" × ⅜" L's
3½" × 3½" × ⅜" L's
3" × 3" × ⅜" L's
¾" dia rivets.
6" × 3½" × ⅜" L
All gusset plates ⅜" thick
All horizontal bars 2¼" × 2¼" × 5/16" L's
All Diagonals 2½" × 2½" × 5/16" L.
¾" dia rivets
6" × 3½" × ⅜" Angle
6" × 3½" × ⅜" Angle
48'-0"
SCALE OF FEET
30
20
10
0
All thig bracing 2¼" × 2¼" × 5/16" Angles - ⅝" dia rivets
DETAIL OF BRACING CONNECTION
Four 1¼" dia holding down bolts in each base
Base angles 6" × 4" × ½" L
Base plate 3'-0" square × ½" thick
centres of feet 18'-6"

Fig. 18

For purposes of description, a trestle has been chosen which would be suitable for supporting the ends of 60 feet span girders, similar to that already described, but carrying a bottom flange track at a height of 48 feet. The arrangement and details are as shown in Fig. 18.

General Considerations.—Adequate clearance for the telpher must be provided for the bogies and main frame of the telpher, and for the swinging load due to wind or the rounding of a curve, all this depending upon the design of machine and nature of the duty. In designing the open trestle there is a temptation to encroach upon this clearance in the endeavour to produce a stiff and pleasing structure. It is fairly obvious that the maximum stress in the legs will occur at the top where it begins to widen out to form the " shoulder," and the wider the legs are made at this point the stiffer the trestle will be. In the design shown, the width at this point is three feet, but it could be made any width up to about five feet without much loss in appearance, and with some gain of rigidity.

In order to gain load clearance and because the bending stresses in the legs are slightly reduced thereby, the bases are placed rather further apart than is usual with the cross-braced trestle.

The leg tapers from three feet wide at the shoulder to two feet wide at the bottom, the taper being a concession to appearance, for a parallel construction would be a little cheaper. Such matters must always be at the discretion of the designer, but in this case there is a reason for not tapering the angles to a point at the bottom, viz., to gain rigidity, for although in the calculations which follow the legs are supposed to be pin-jointed to their foundations, the bases well bolted down (perhaps concreted up solid) will certainly have a considerable degree of fixity, which increases the rigidity of the trestle to an even greater extent than it assists the strength.

The girders are bolted at their ends to the faces of the trestle and their weight is transmitted to the trestle by these bolts in shear. If the trestle top framing is made the same depth as the girders a cover plate can be placed right across the top connecting the trestle and the two girders, but in doing this it should be remembered that the girders can no longer act independently as simply beams. Girders of the Warren type may be connected, by their bottom booms only, to a trestle of similar design to that shown.

Calculations.—The frame and stress diagrams are shown in Fig. 19. The wind and telpher loads are taken from the 60-foot span girder already described, and the dead weight of the trestle is ignored, as it is confusing to the diagram to include a series of vertical loads acting nearly parallel to the main members from top to bottom, and as they are direct loads and do not induce any loads greater than their own value they are easiest added afterwards.

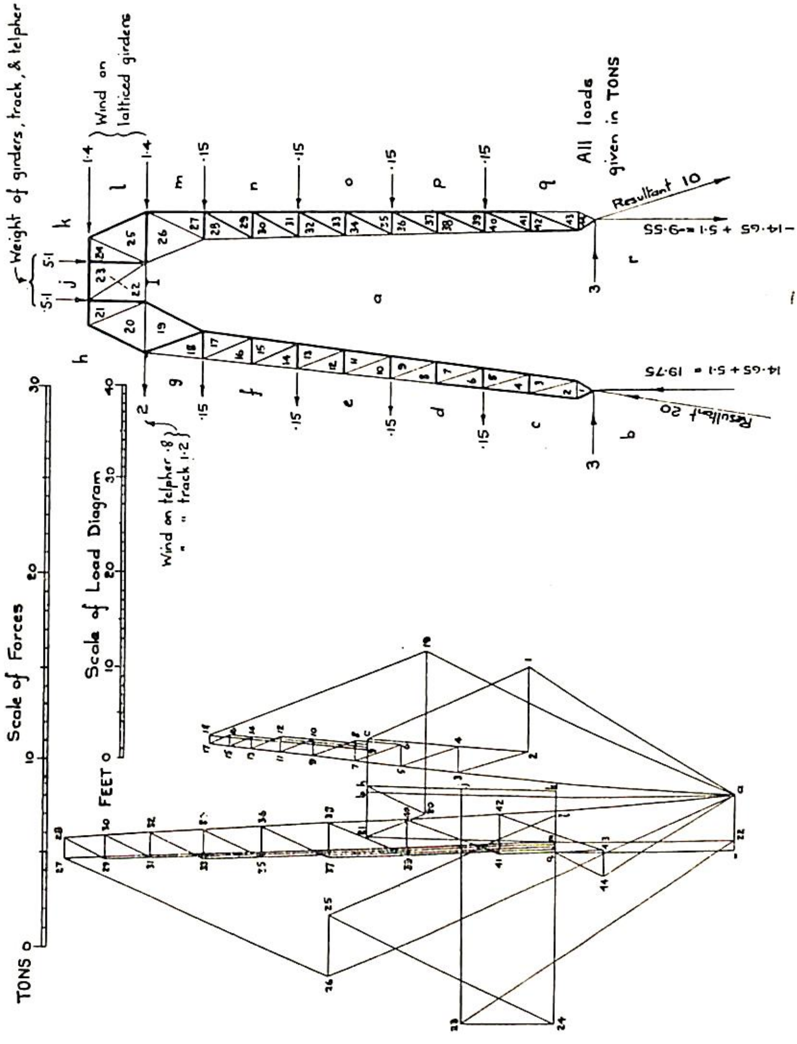

Weight of girders, track, & telpher
5·1
5·1
Wind on latticed girders
1·4
1·4
·15
Wind on telpher ·8
" " track 1·2
2
All loads given in TONS
Resultant 10
Resultant 20
14·65 + 5·1 = 19·75
−14·65 + 5·1 = −9·55
3
3
Scale of Forces
TONS 0 10 20 30
Scale of Load Diagram
FEET 0 10 20 30 40

Fig. 19

Referring to the frame diagram, it will be noticed that the total horizontal wind load of 6 tons is resisted equally by the two legs, and whether this is justifiable is a debatable point. Readers will be familiar with the argument usually advanced against dividing equally between the two shoes the horizontal forces on a roof truss, and the argument is quite convincing. Since there is some remote resemblance between the open trestle and a roof truss it is reasonable to question the argument to see if it applies equally to the open trestle. The writer's opinion is that, due to its comparatively great flexibility, the open trestle will share the horizontal load much more nearly equally between its two foundations than is likely to be the case with a roof truss, which is frequently mounted on supports of uncertain relative rigidity in a horizontal direction.

Virtual bars C1, A1, A44, and Q44 are used on the diagram, in order to complete the figure at the feet.

The assumption of pin joints at the base level is a rather severe one, as the anchoring effect of the feet will certainly introduce in each leg a point of contraflexure some distance up from the base, thus reducing stresses in all main leg members, though not appreciably affecting the bracing members of the legs, as the horizontal shear forces remain the same.

The wind on the trestle itself is shown by the small loads of ·15 ton each, applied at every second panel point. That further subdivision of this load into ·075 ton at *every* panel point is needlessly accurate may be seen by reference to the stress diagram, where the points *d e f* and *n o p* are so close together that there is no room for the letters.

The vertical reactions at the bases due to overturning moment are shown, but the calculation is not given, as the method is the same as that already described for the ordinary trestle.

The two resultants at the bases have also been drawn on the frame diagram, but are omitted from the stress diagram, as they are of no special importance.

No description is given of the method of construction of the diagram, for if the reader is not already familiar with the process he will not learn it from an apparently complicated diagram such as this, but he may be assured that the actual making of a stress diagram is among the least difficult parts of a designer's work, for it is a purely mechanical operation, requiring only close attention and a careful pencil.

Table I. gives the loads in some of the principal members, the sections used, the resultant stresses, and the number of rivets. Note that the loads in Column 2 are half the value scaled from the diagram, as each load is shared by two bars.

TABLE I.

Member.	Load from diagram.	Section used.	L/R.	Nett area.	Perm. Stress.	Act. Stress.	No. of Rivets.	Value Rivets.	Remarks.
a. 17	**14·3**	$6'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$ L	79	3·42	4·35	4·17	9—$\frac{3}{4}''$	18	
m. 27	**13**	$6'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$ L	79	3·42	4·35	3·8	7—$\frac{3}{4}''$	14	
a. 28	17·8	$6'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$ L		2·63	7·0	6·8	9—$\frac{3}{4}''$	18	(two $\frac{3}{4}''$ & one $\frac{5}{8}''$ rivet deducted.
l. 25	**6·75**	$3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$L	106	2·49	3·65	2·7	4—$\frac{3}{4}''$	8	
18·19	**6·25**	$3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$L	108	2·49	3·6	2·5	4—$\frac{3}{4}''$	8	
24·25	6·75	$3'' \times 3'' \times \frac{3}{8}''$L		1·55	7·0	4·35	4—$\frac{3}{4}''$	8	
42·43	**3**	$2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}''$L	125	1·47	3·1	2·04	2—$\frac{3}{4}''$	4	

(The figures in heavy type indicate that the load is compressive).

On the diagram, members in compression are shown in heavy lines, but of course a great many members sustain a reversal of stress when the wind blows from the opposite direction. Some members are struts for all conditions of loading, and it is probable that 20/21 and 24/25 are ties under any conditions. Corresponding members must be designed according to which is the worst condition, and a case may occur where a member on one side is in compression under a certain load, while the corresponding member on the other side is in tension under a greater load. In such a case the size of the member might require to be selected according to its value as a strut, but the number of rivets in the connection would necessarily be determined according to the load it carried when in tension.

This question of reversal of stress raises the question of limiting the range of permissible stress.

In comparing the nature of the reversal with that occurring in such cases as the journals of railway wagon axles, where such limitation is usual, it will be seen that the conditions are so dissimilar that no comparison can be made. The railway wagon axle is definitely subjected to high speed reversals of bending stress of more or less known magnitude, and these stresses approach the safe limit. In the case of the open trestle the stresses in either direction depend mainly upon the wind, which may reach the assumed maximum velocity in the assumed direction once or twice in the lifetime of the structure. Normally it will be subjected to reversal once every few days of a magnitude much less than a tenth of the maximum. Such conditions do not justify any stress limitation due to reversal, and the wind bracing bars may be designed for the safe maximum stress in tension and compression.

Referring to the general design (Fig. 18), it will be seen that the width of the legs, in the direction of the track, has been made two feet parallel, braced with $2\frac{1}{4}'' \times 2\frac{1}{4}'' \times \frac{5}{16}''$ angles. Since the ratio of slenderness is only about 25, the stability of the leg as a column in this direction will not need investigation.

The calculated weight of the trestle is $4\frac{1}{2}$ tons, which corresponds to approximately ·083 tons per foot of height. This weight is taken almost entirely by the eight main leg angles in compression, and it will be seen that as the combined area of the leg angles is 22·5 square inches, the added stress even at the lower ends of the legs is only ·165 tons per square inch. It is therefore not necessary to investigate the stress due to trestle weight unless the section selected is already stressed very closely to the safe limit.

One further point may be mentioned; equal angles have been used for the bracing, and these are connected at their ends by one flange only, thus causing the angle to be eccentrically loaded, which may be of some importance when it is acting as a strut. Some designers use unequal angles in such cases and put the rivets in the broader flange, thus reducing the eccentricity (but the value as a strut is also reduced, though perhaps not in the same proportion).

LONGITUDINAL FORCES.

Forces acting on the structure along the direction of the track are due to either wind or the accelerating forces consequent on starting and braking of the telpher machine. The wind force may be a very considerable factor.

To estimate accurately the longitudinal wind resistance of latticed box girders for a given wind velocity is clearly impossible. We shall be fortunate if our estimate is within 10 per cent. of the truth, for we do not even know at what angle to the line of girders the wind must blow in order to produce the maximum longitudinal component, neither can we do anything but guess at the effective exposed area, for the eddy currents produced by the bracings first met by the wind must have the effect of reducing in a very uncertain manner the effective mean velocity for leeward or semi-masked bars. Probably the angle which produces the maximum longitudinal component is about 30° with the track line, as shown in Fig. 20, and then the components of the wind pressure are :—

Longitudinal component $=30 \times \cos 30^\circ = 26$ lbs. per square foot.
Broadside component $=30 \times \sin 30^\circ = 15$ lbs. per square foot.
(but see footnote below).

The writer's practice is to disregard skin frictional wind resistance, and thus the longitudinal boom members and the runway joist are taken as offering no resistance, neither is any allowance made for projecting rivet heads and boltheads. The wind is assumed to act at the full resolved pressure on the nett area of all transverse bracing bars on the windward side of the girder and on the top side. Half this pressure is taken as acting on the leeward side bracings and the bottom bracings. This is a purely empirical method, but it gives results which appear reasonable without being excessive.

It is doubtful whether the assumptions made above are correct, for the force to be resisted is not applied through a rigid body as in ordinary statics. Instead of resolving the wind *pressure* into two components, it would probably be more nearly correct to resolve the wind *velocity* into two components, and since the wind pressure varies as the square of the velocity the pressure values are—

Longitudinal component $= 30 \times \cos^2 30^\circ$
$= 30 \times \cdot 75 = 22\cdot 5$ lbs. per sq. ft.
Broadside component $= 30 \times \sin^2 30^\circ$
$= 30 \times \cdot 25 = 7\cdot 5$ lbs. per sq. ft.

It will be noticed that these figures are considerably less than those obtained by resolving the pressure, especially the broadside component, which for an angle of 30° is halved. The whole question of wind forces on latticed structures is very complicated and does not lend itself to computation from irst principles.

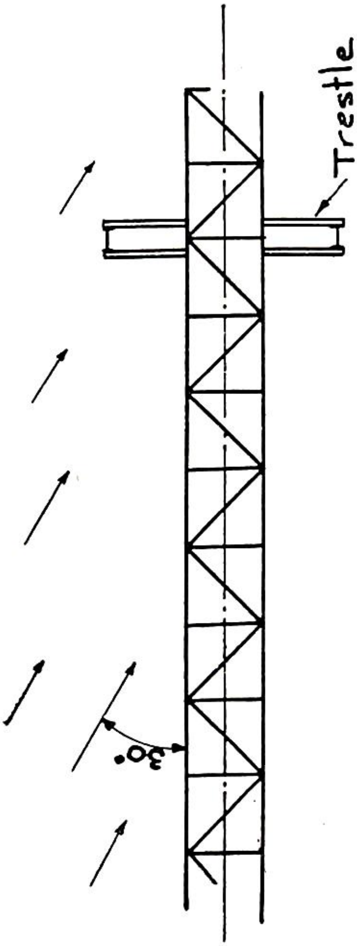


Fig. 20

For a box girder 70 feet long, seven feet deep and four feet wide, with ten panels of bracing composed of 3″ × 3″ angles (average size), the longitudinal force works out at about 1¼ tons. The question of resisting this force is largely influenced by the general layout of the structure. Sometimes, as is frequently the case with tracks containing several curves, the provision must be made more by judgment and experience than by calculation.

Fig. 21 shows a plan of the telpher structure* illustrated in Fig. 1. It will be seen that stability in all directions is obtained by so placing the trestles that longitudinal forces are in every case resisted by a suitably-placed trestle.

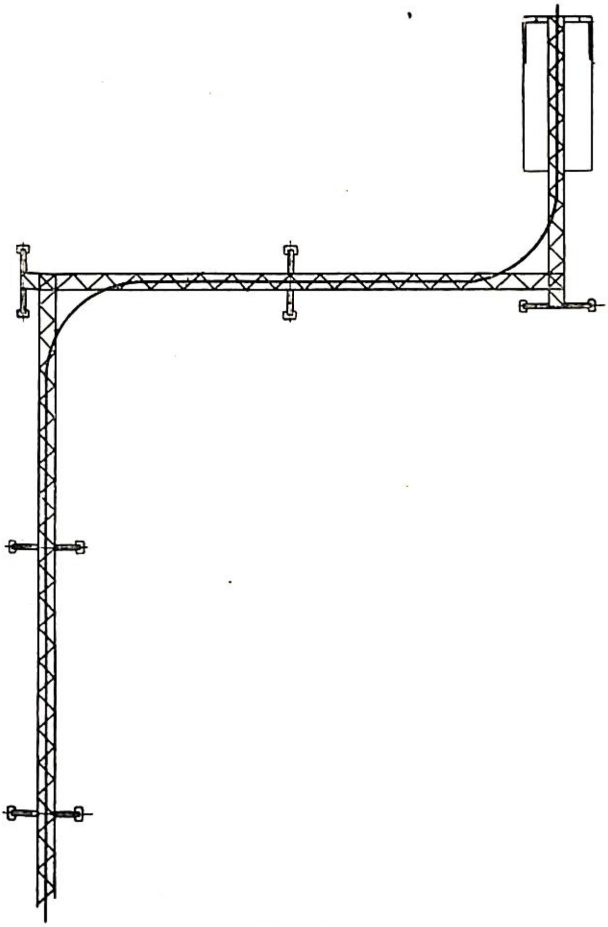


Fig. 21

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*By Messrs. Strachan & Henshaw, Ltd., Bristol.

The writer has seen telpher structures which showed no evidence that wind forces in all directions had been allowed for, and he is of the opinion that many of them could not without damage sustain a uniform wind pressure of 30 lbs. per square foot *in their weakest direction.* The comparative rareness of such wind pressures in this country, combined with the fact that telpher structures are frequently in semi-sheltered positions (*i.e.* the wind has to traverse obstructed ground before reaching them), explains the freedom from trouble.

Longitudinal wind forces are sometimes resisted by knee struts connecting the lattice girder to the legs of the trestle, as shown in Fig. 22.

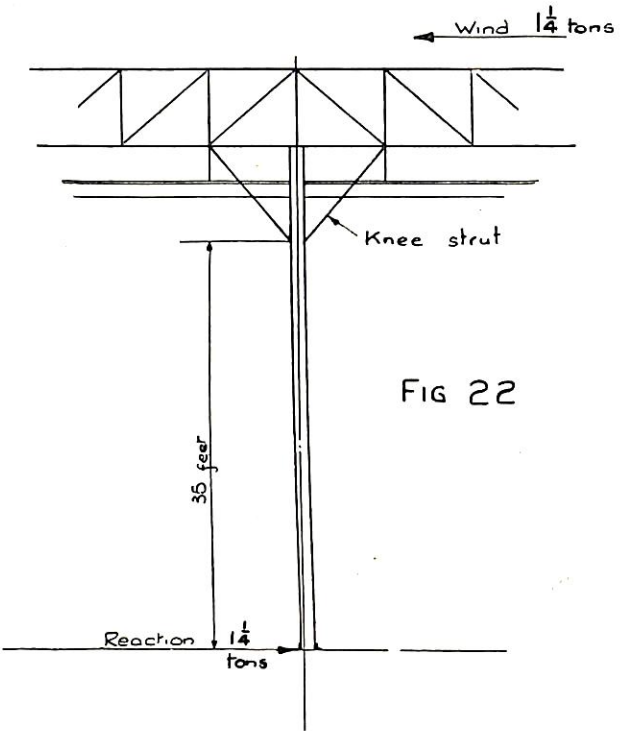

FIG 22

In this case a considerable bending moment is introduced into the trestle leg and an additional force is added to the girder.

For the case shown in Fig. 22, the bending moment in the legs is,

1·25 tons × 420 inches = 525 inch tons.
which, for one leg = 262 inch tons.

For a bending stress of 4 tons per sq. inch (the leg is carrying other loads)—

$$Z \text{ required} = \frac{262}{4} = 65{\cdot}5$$

A 14″ × 6″ R.S.J. has $Z = 62{\cdot}9$

But consider the deflection (top end fixed, lower end hinged)—

$$\text{Deflection in inches} = \frac{W\,L^3}{3\,E\,I} \text{ (approximately).}$$

$$= \frac{1{\cdot}25 \times 420^3}{3 \times 12{,}000 \times 440} = 5{\cdot}85 \text{ inches.}$$

Therefore the vertical reactionary load at the trestle base is applied to the trestle leg with an eccentricity of nearly 6″, which figure will not, however, be realised in practice, as the trestle base certainly possesses some degree of fixity, which will reduce the bending stress, and to a still greater extent the deflection. The deflection is, however, the main objection to relying solely on knee struts for resisting the longitudinal forces, for lack of rigidity is noticeable with much smaller forces than those which would be produced by a 30 lbs. per square foot wind pressure, and will probably reveal itself as soon as the plant begins to work.

If the structure consists of a long range of girders carrying a track which is practically in one straight line, some definite provision against endwise forces is essential, and if no independent anchorage is available the best way is to substitute one or more substantially braced towers in the place of trestles.

Ordinary trestles should be taken as having no fixity (longitudinally) at the foundations, hence nearly half the total windage (in the direction of the track) acting on the faces of the trestles is transmitted to the girder connections at the top of the trestle, and thence through the range of girders to the main anchorage.

Forces due to Acceleration.—Longitudinal forces due to rapid acceleration and braking of the machine do not present much difficulty. As a rule one of the two bogies is driven, and braking is done on the same bogie. Assuming that the weight of the machine is equally shared by both bogies and that the coefficient of friction between rail and wheel treads is ·2, the maximum possible tangential force will be $\frac{1}{10}$ of the weight of the machine, as skidding will occur if anything more be attempted.*

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* This is not absolutely correct, as the braking is necessarily *less* effective when the driving bogie is leading than when it is trailing, and *more* effective when the machine is travelling in the reverse direction, this being due to the

The minimum distance in which the machine can be accelerated from rest to full speed or brought to rest from full speed is easily calculated, and is independent of the weight of the machine for a given distribution of loads on bogies and coefficient of friction. Let these be as mentioned above and the maximum speed of the machine be 600 feet per minute, *i.e.* 10 feet per second, then

Kinetic energy = work done in accelerating from rest,

$$i.e., \quad \frac{Wv^2}{2g} = \frac{W}{10} \times \text{distance},$$

$$\therefore \quad \text{distance} = \frac{10v^2}{2g} = \frac{10 \times 100}{64{\cdot}4} = 15{\cdot}5 \text{ feet}.$$

This corresponds to an acceleration of 3·2 feet per sec.^{2}, a result which can be obtained in braking, but telphers are rarely fitted with motors capable of producing an acceleration greater than 2 feet per sec.^{2}. (See page 9).

(*Note.*—The London Underground Railways claim an acceleration of 2·2 feet per sec.^{2}).

Centrifugal Forces, due to the machine rounding the curve, must sometimes be considered. In general, a 20-feet radius curve will be taken at about half full speed, say 300 feet per minute, *i.e.*, 5 feet per second.

For a machine, complete with load, weighing 6 tons,

$$\text{Centrifugal force} = \frac{Wv^2}{g\,r} = \frac{6 \times 5 \times 5}{32{\cdot}2 \times 20} = {\cdot}23 \text{ ton}.$$

The force due to this cause is therefore relatively small.

One more longitudinal force remains to be mentioned, viz., wind pressure on the end of the telpher machine. The interest in this lies not so much in the force transmitted to the structure, but whether the machine is likely to be blown along the track.

The endwise projected area is about 30 square feet, giving, at 30 lbs. per square foot, a total pressure of 900 lbs. If the machine without load weighs 8000 lbs. and the effective coefficient of friction be taken as before at ·1, the resistance is only 8000 × ·1 = 800 lbs., and the machine would thus be blown along the track even with the brakes hard on, but the writer is not aware of any case where this has happened, which tends to confirm that such wind loads are rare.

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horizontal couple set up by the tangential force at the rail and the opposite accelerating force at the centre of mass several feet below. This horizontal couple must be balanced by an equal and opposite couple applied to the bogies, hence the reactions of the bogies on the rails adjust themselves accordingly. The exact amount by which the effective bogie loads are altered is easily calculated if the position vertically of the centre of mass is known.

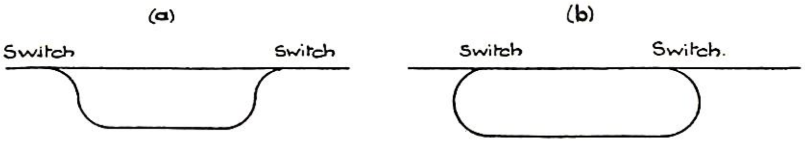
Anchor chains for attaching the telphers to the structure were provided in one case for a plant in India, where the wind pressure was specified at 50 lbs. per square foot.

INCLINED TRACKS.

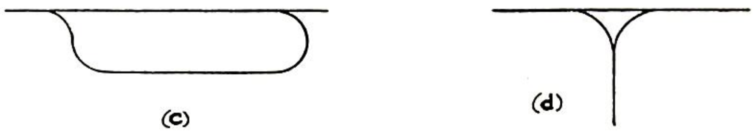
It is quite practicable to arrange for the whole or a portion of the track to be inclined, but this is not often done, as the maximum reasonable inclination is about 1 in 40, so that for tracks of average length the possible extra headroom obtainable is only a few feet, and the construction, especially on the curves, is more difficult. The factor which limits the inclination of a short gradient is the frictional grip between the rails and the wheels in wet weather, and this is as important in descending as in ascending, if it is required to slow down or stop on the incline. Sheltering the rail from direct rain has been tried, but the value of this is doubtful unless it can be carried along the whole track so that the wheels can be kept dry. A telpher with all its wheels driven should do better on inclines.

SWITCHES.

A branch track may be switched off at almost any part of a telpher track, and on either side whether the system be top rail or bottom flange type. These branch tracks can be looped back into the main line again if desired, and without restriction as to change of direction in the case of the bottom flange system, but with the top rail system change of direction is not possible owing to the non-symmetry of the track and telpher. Fig. 23 illustrates these limitations.



Suitable for either top rail or bottom flange systems.



Suitable for bottom flange system, but impossible with top rail system

Fig. 23

Although looping back as in (*c*) and (*d*) (Fig. 23) is mechanically possible with telphers which run symmetrically on the bottom flange of a joist, the change of polarity of the current supply must be provided for, unless the system is single wire with earth return.

It is usual to mechanically connect to the track switch a trolley wire switchpoint, which not only serves to deflect the trolley wheel on to the correct line, but also renders electrically " dead " a considerable length of the unconnected portion of the line. This is important when more than one machine is operating on the same track system, as it prevents a second machine from travelling into and fouling a switchpoint which is set against it. The dead section should be as long as possible, consistent with leaving the second machine a long enough live portion to work on. A section insulator, similar to those used in tramway work, is fitted in the trolley line between the dead and live portions, and the live portion is either independently fed or is bridged across the dead section from the other side of the switch.

There are many types of track switch, but there is not room to describe them here. The operation of a switch is usually arranged so that no one but the driver of a telpher can reach the operating point, which is usually situated on a small platform attached to a trestle near the switch, and on to which the telpher driver can easily step from his cabin.

STORAGE OF MATERIALS OVER A LARGE AREA.

The top rail telpherage system can readily be adapted to conditions where it is required to distribute material to an even depth over a rectangular area. Storage nearly always involves reclaiming, and it is an advantage when the same plant which does the storing can as easily discharge the store.

As shown in Fig. 24, a branch track at right angles to the main track is provided with a running switch, and the whole of the curve and branch track is carried by a girder which completely spans the store.

The switch end of the girder is carried by a travelling under carriage, which runs on a track spanning the trestles of the main telpher track. The outer end of the bridge is supported by a travelling frame which runs on a rail near ground level, and this frame is usually pivotted to the girder.*

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* An exception is the travelling bridge (by the Mitchell Conveyor and Transporter Co.) at the Clyde's Mill Station of the Clyde Valley Power Co., where the outer leg is fixed to the girder and the endwise variations are taken up by a floating arrangement at the switch end. The reasons are too involved to be given here. (See " Modern Telpherage and Ropeways," Blyth).

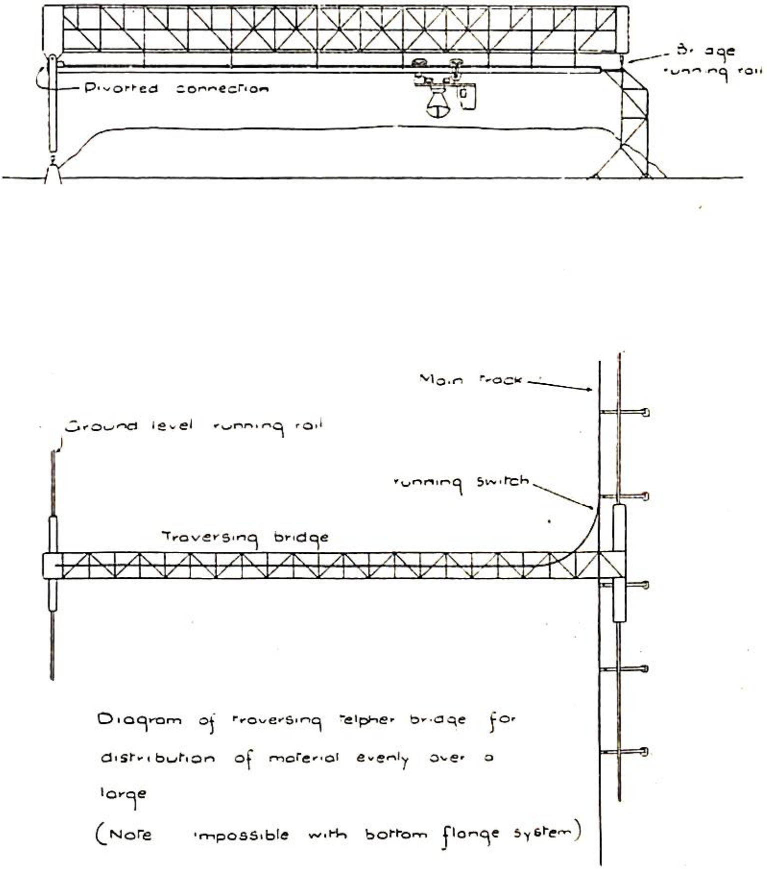
Bridge running rail
Pivoted connection
Main track
Ground level running rail
running switch
Traversing bridge
Diagram of traversing telpher bridge for distribution of material evenly over a large
(Note impossible with bottom flange system)

Fig. 24

Fig. 24 shows the simplest case, but there are many possible variations, for example, the coal handling plant at Clyde's Mill in which the travelling bridge carries two tracks, one on either side, the two being looped together at the outer end and both connected by running switches to the main track. The telphers are therefore able to pass the bridge, which is not possible in the case illustrated. A still further example is the coal storage plant at Messrs. Nobel's explosive works at Pembry,* where again the bridge carries two tracks, not connected at the outer end, but both connected by running switches to the main track, and is provided with a special shunt line, which enables the telphers to pass the bridge without having to travel to the outer end.

The bridge must be regarded as a semi-fixture, as it is necessarily a heavy structure, and it is not practicable to move the bridge for every trip made by the telpher, neither is this necessary. The travelling speed is therefore made fairly low, about 30 feet per minute. The travelling gear, which is driven by one or more electric motors, is situated either as a single unit near the centre of the bridge, or in two inter-connected units, one mounted on each undercarriage. It is absolutely necessary for both ends of the bridge to be driven, though not necessarily all the wheels. It should be remembered when designing the travelling gear that the bridge must be capable of travelling against a moderate breeze.

It is hardly necessary to point out that the traversing bridge cannot be applied to the bottom flange system, the running switch being the difficulty.

GRABS.

Any type of Grab that can be carried by a crane can be carried by a telpher, but in addition to grabs of the single-chain and single-rope type, grabs having two ropes or three ropes may very conveniently be used, and are usually preferred, as the grab is thereby prevented from spinning, which is a useful property when it is required to operate in confined spaces, such as railway wagons and barges. Grabs are often discharged, as in crane practice, either by resting or by tripping when in the top position, but there are several makes of grab which can be gently discharged by the driver at any height without resting, the three-rope grab being one of these.

SKIPS.

Skips of many types, shapes, and sizes have been used on telphers to meet the special requirements of each plant, but they may be roughly divided into two general classes, viz., tipping skips and skips which are provided with trip doors. For carrying red-hot coke direct from horizontal retorts in a gas works the self-tipping and self-righting type of skip is nearly always used, and is built up from mild steel flats and angles spaced apart to facilitate quenching when the skip full of hot coke is lowered into the water quenching tank. Cold coke from vertical retorts is usually carried in plate skips, and sometimes these are of the tipping type. Occasionally it is convenient to carry coal in tipping skips.

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* By Messrs. Strachan & Henshaw Ltd., Bristol. See "Engineering," (11-7-1919), and "Modern Telpherage and Ropeways," Blyth.

Fig. 2 is an illustration of a typical gasworks telpher, carrying a self-tipping and self-righting hot coke skip.

The proportions of this type of skip are so arranged that the combined centre of gravity of the skip and its contents is above the trunnions when the skip is loaded, and that the centre of gravity of the empty skip is below the trunnions. If the distances of these respective centres of gravity are far enough above and below the centre of rotation to overcome the frictional resistance of the trunnions, the tipping and righting is quite satisfactory.

When loaded the skip is kept in the upright position by means of a catch attached to the bridle, or bale, and attached to this catch is a cord which passes through the driver's cabin and is kept just taut by a suitable take-up device. To discharge the skip at any height the driver pulls the cord, which releases the catch and allows the skip to over-balance.

The writer wishes to acknowledge his thanks to Messrs. Strachan & Henshaw, Ltd., Bristol, for their kindness in lending the blocks for Figures Nos. 1 and 2.